# THE HEATING AND VENTILATING MAGAZINE

VOL. VI., No. 1 NEW YORK, JANUARY, 1909 $1.00 a year, 10 cents a copy

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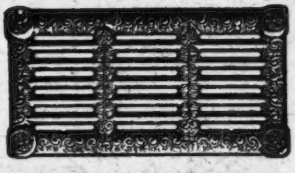

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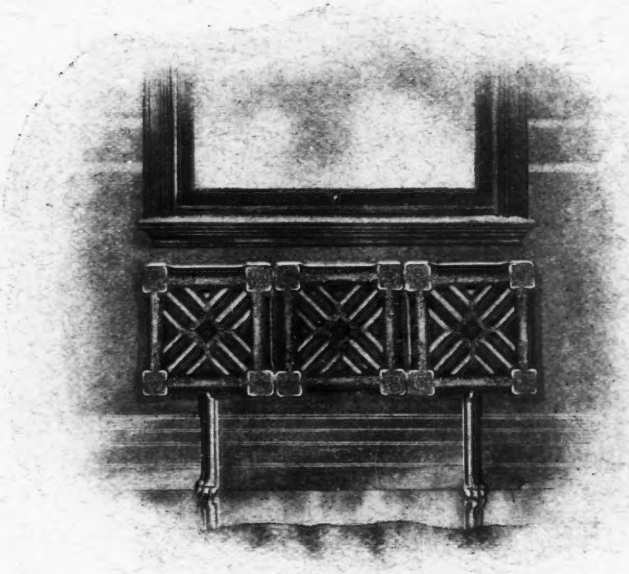

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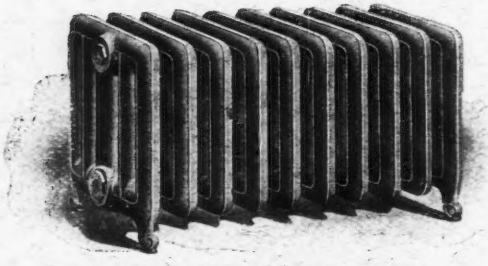

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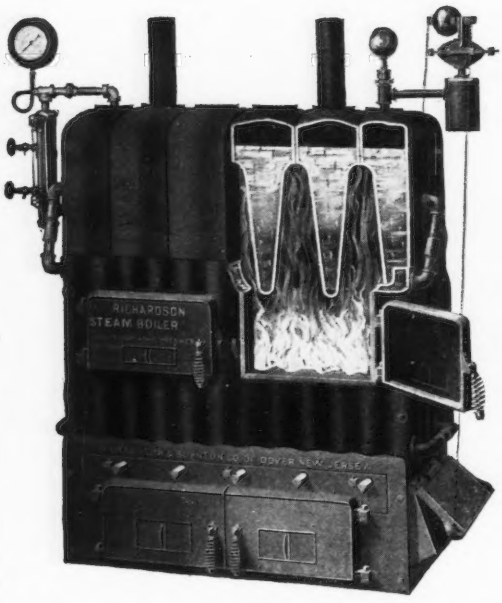


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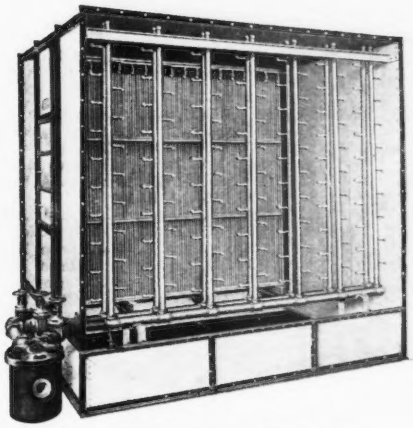

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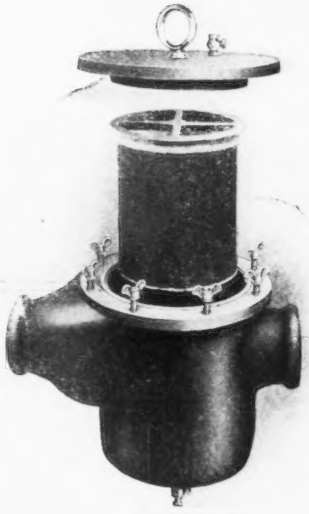

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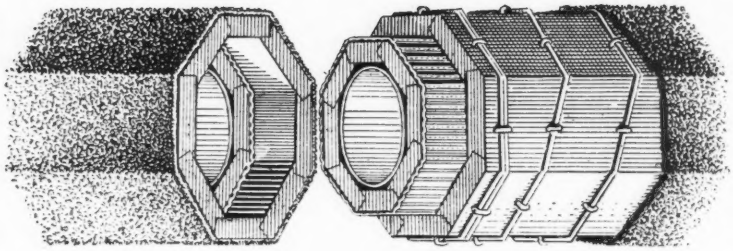

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VOL. VI No. 1


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THE HEATING AND VENTILATING MAGAZINE

1123 BROADWAY NEW YORK JANUARY, 1909

# Systems of Shop Heating

By F. R. STILL*

What system of heating is best adapted for the shop under consideration is frequently an important question. There are all kinds of methods—some good, many poor, and a large number positively bad. It is not within the province of this article to analyze the merits and demerits of the various systems, but to point out the advantages obtainable with a good system of heating and to explain how it should be laid out, installed and operated to secure the best results.

Those who recognize the advantages of a good heating plant frequently do not realize the necessity for designing it when the plans of the building are being made. The result is that instead of arranging other things of much less importance to accommodate the heating plant, the latter has to be contorted, cut down or squeezed into whatever space is available after everything else is taken care of. This makes the plant either expensive to install or to operate, and often is so contorted or dwarfed in one way or another as to fail to give the satisfactory results naturally expected.

## THE CHARACTER OF THE BUILDING AN IMPORTANT FACTOR

When it comes to laying out a heating plant, the character of the building and the purpose for which it is built must be carefully studied. What answers for one type of building is totally unsuited to another of a different type. Again, what serves very nicely in a building of a certain type, in which a particular class of work is done, proves anything but satisfactory in a building of the same type, in which a different kind of work is done.

This can be made clear by comparing a machine shop with a paper mill. They are similar in size, shape and exposure. In the machine shop, no steam or moisture is emitted to condense on the walls and roof; hence many of them are heated in a fairly satisfactory manner by direct radiation, consisting usually of steam coils strung around the outside walls. Such a system would never do in a paper mill, where every day several tons of water are thrown off into the atmosphere in the form of steam from the drying cylinders over which the paper passes. This steam must be taken up by a copious circulation of warm, dry air, or it will drip from the roof and run down the walls in streams.

Again, in a shoe factory, shirt, glove, cap or other manufacturing plant, where hundreds of employes are assembled in one room, the air very soon becomes unfit to breathe; it produces a dull, languid feeling, which saps all the vitality and ambition of the employees, makes them inert, forgetful, careless, subject to severe colds and other disabilities, all tending to make them irregular in their attendance to their duties. To overcome this, the building must be well ventilated, and the ventilation of a building in the

*Chief engineer, American Blower Co., Detroit, Mich.

winter time must be considered in connection with the heating system.

Foundries are much like the paper mills just cited, as the steam rising from the moulds when pouring off, as well as the smoke and noxious gases, should be removed. Roof ventilators, monitor lights or ventilating sashes are very poor makeshifts indeed. Perhaps they will let some of the smoke escape and with it some of the gases and most of the heat; but most of the gases, being heavier than the air, lie in the lower strata near the floor and are not affected by the roof vents. Further, instead of removing the steam, they only accelerate the annoyance of it, as the warm air which escapes through the roof must be supplanted by outside air, and this, not being properly heated, chills the air within the building, thus reducing its capacity to carry moisture, thereby causing greater precipitation than there would be were no attempt made to ventilate the building. Instances are numerous where so much moisture has been condensed on the roof as to spoil the moulds on the floor, and this even in foundries well heated, so far as temperature is concerned, by direct radiation.

Then again, apart from the use to which the building is put, consideration must be given to its character.

The tendency nowadays is to put up lofty buildings. Such structures have steel frames, walls made almost entirely of glass, with only narrow pilasters between, and shallow panels of brick or concrete above and below the openings. The exposures are relatively enormous, requiring large amounts of heating surface, yet the spaces available for steam coils are limited to such an extent that they are often insufficient for the heating surface required.

Furthermore, when the space is sufficient, the banking of such great heating surfaces into a small space, makes it uncomfortable for the employees who have to work near them.

### FLEXIBILITY OF BLOWER SYSTEM

It is these and many other features that have made the blower system popular, as it has proven efficacious in almost every building of whatever character, when it has not been misapplied. Many of the designers of such heating plants seem to have a penchant for trying to make this system do all sorts of unusual and almost impossible stunts. The system is so flexible and so positive in its action that many persons seem to think it capable of defying Nature's laws. But occasionally Nature asserts her rights in a most emphatic manner, with the result that a plant fails to do what was expected of it. This, however, is no fault of the system, but rather of the man who designed the plant.

The blower system, sometimes called hot blast system, plenum system, or mechanical heating and ventilating system, is so universally known as to need no description of the apparatus, though there is plenty to be learned about the best way to install it.

The ducts are usually made of galvanized sheet iron for all types of buildings, except where it is necessary to place them under ground, in which case they are built of brick, vitrified tile or reinforced concrete.

It is always advisable to run the ducts inside the building because of the heat wasted if they are run outside, be they above or below the ground. It is, furthermore, advisable to make the ducts of galvanized iron and suspend them in the air overhead, if possible, as the loss of heat from underground ducts is enormous.

### LOSS OF HEAT FROM UNDERGROUND DUCTS

A fair idea of what this loss sometimes amounts to may be gained from mention of a case a few miles north of Grands Rapids, Mich., at a large silk mill. The basement wall is built of stone about 3 ft. thick. The duct is built of brick above the basement floor, along the outside wall, beneath the windows.

In the winter when the outside temperature is down near zero and there is a foot or more of snow on the ground, the grass is perfectly green

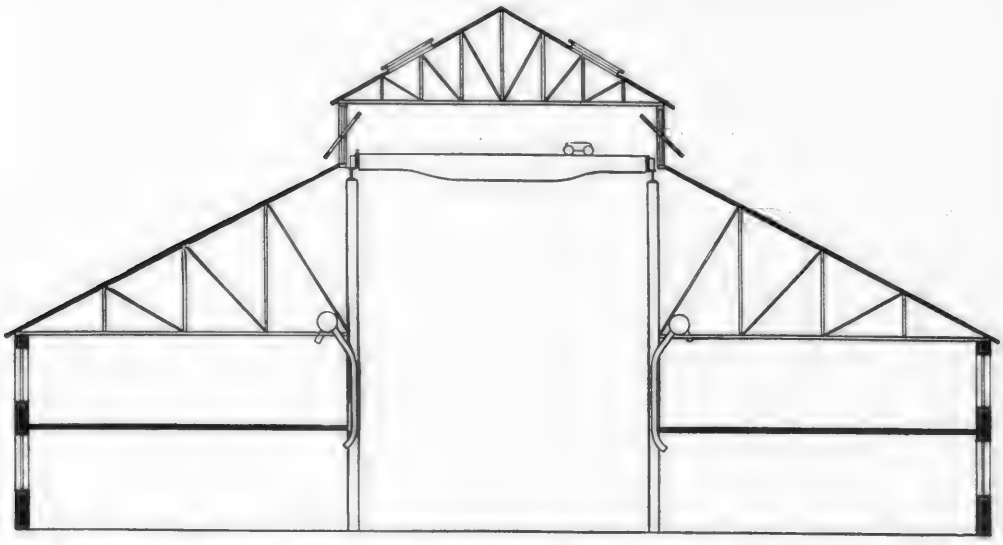


FIG. 1.—METHOD OF TAKING BRANCHES FROM MAIN HOT AIR DUCTS

for a space extending out from the building nearly 10 ft. in width, and for 10 ft. more there is no snow, though the grass is dead.

Had the duct been made of iron and run inside the building, all this heat would have been radiated into the surrounding atmosphere, thus helping to maintain the required temperature, instead of keeping the grass green. The location of trunk line ducts and the disposition of the branches therefrom is a matter requiring considerable practical experience and good judgment.

CONSTRUCTION OF MODERN SHOP

The modern shop has a trussed roof, the bottom chords of which are often 30 to 50 ft. above the floor. The ducts have to rest on these trusses and it is impossible to drive air from them down to the floor in a way that will produce satisfactory results. This method has been tried often enough, but there still remain others who must be convinced of its impracticability by trying it themselves. The only way to obtain an even distribution of heat, is to discharge heated air at such points as it is most needed and where the effect will be most appreciated.

To distribute the heat evenly necessitates running the ducts to all cold spots; it is needed the most in the lower strata near the floor, not up among the roof trusses; the greatest benefit is derived from the system by diffusing the warm air close to the floor, keeping the lower strata in cir-

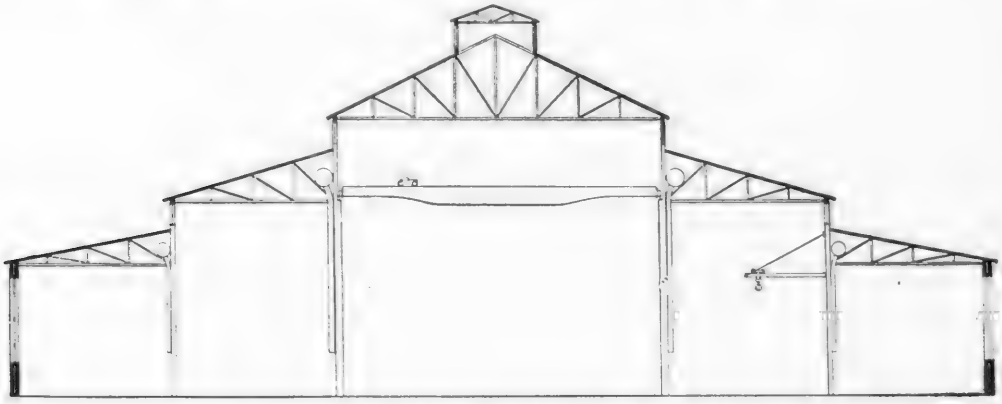


FIG. 2.- ANOTHER METHOD OF TAKING BRANCHES FROM MAIN HOT AIR DUCTS

culation and thereby warming it by mixing with it the warm air discharged from the ducts.

The best way to bring this about is to extend the branch ducts from the main trunk line over to the walls or to posts not more than 20 ft. away from the outside walls, then down toward the floor, ending 4 or 5 ft. from the floor. (Figs. 1 and 2.) The air should discharge directly toward the floor or at only a slight angle from perpendicular. This method will be found most effective in machine shops, foundries and other lofty structures.

In paper mills, rubber works, dye houses and other plants for which the building is of the same type as those just noted, it is necessary to blow some hot air out towards the roof as well as down towards the floor (Fig. 3), in order to take care of the condensation which would otherwise collect on the under side of the roof. Even then, in very cold climates, it is sometimes necessary to put in a false ceiling to overcome this annoyance, particularly if the roof is built of a material which is a good conductor of heat.

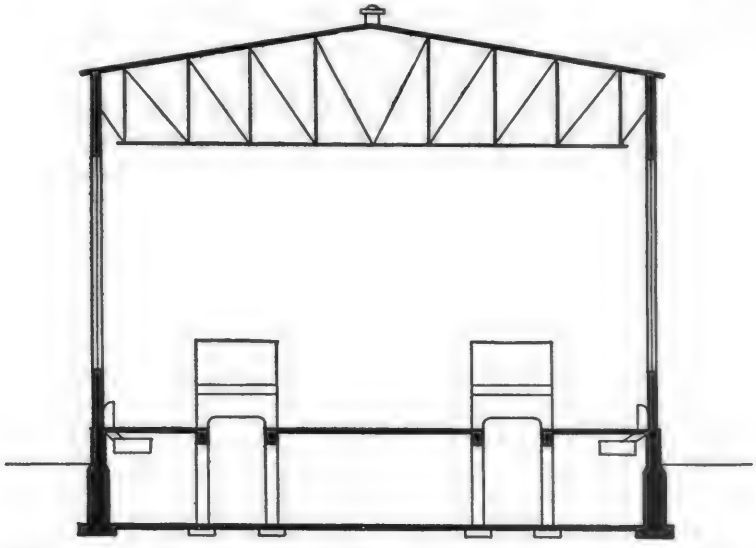


FIG. 3.—DESIGN FOR SHOP HEATING SYSTEM TO AVOID CONDENSATION ON UNDER SIDE OF ROOF

For buildings which are several stories in height, each story being from 10 to 16 ft. high, the treatment should be different. With them, it is possible and sometimes advisable, to introduce the air near the ceiling, blowing downward at an angle of 30° to 45° from horizontal.

Frequently buildings of this character are quite effectively heated from one or two galvanized iron stand-pipes run up through the middle of the building with outlets into each story (Fig. 4). This method is practical in buildings not over 60 ft. wide; if the building is not over 100 ft. long, one riser will be sufficient.

For cotton, woolen, or silk mills, it has become almost the universal practice to build vertical warm air flues on the outside face of the pilasters, on both sides of the building (Fig. 5). These flues usually have a 2-in. air space built into the brick work to insulate them. The air is admitted to each story about 8 ft. above the floor. Deflecting "Mill Dampers" regulate the volume of air discharged through each opening. The various flues can be supplied at their base from a main duct built either of masonry or galvanized iron.

For manufacturing plants, it is customary to make the trunk line ducts of such an area as will convey the required volume of air at a velocity varying from 1,500 to 2,400 ft. per minute. In high buildings used for

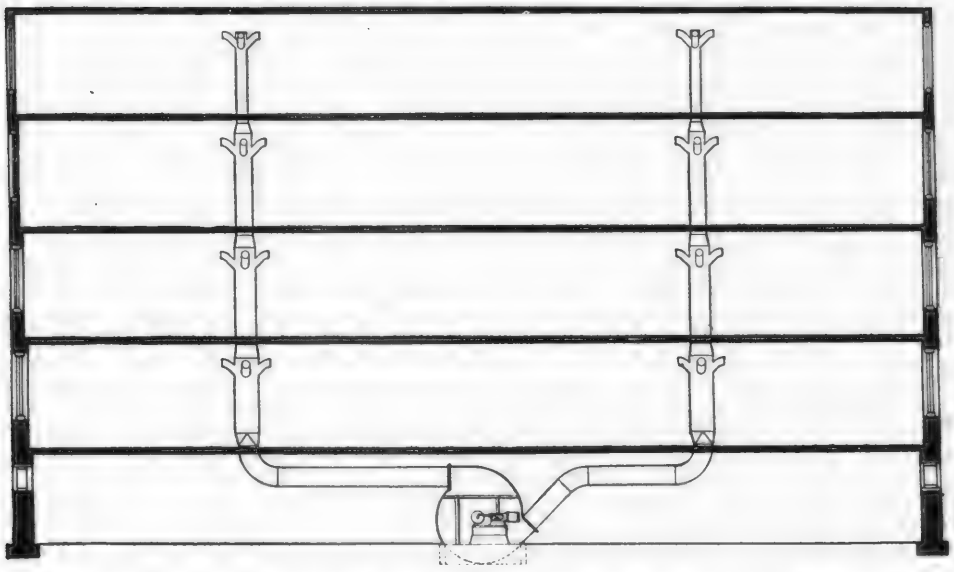


FIG. 4.—SHOP HEATING SYSTEM WITH STAND-PIPES HAVING OUTLETS IN EACH STORY

heavy and coarse work, where most of the employees stand or move about considerably, the velocity can be much higher than in shops divided into several stories, or those in which the work is more or less sedentary, like the manufacture of shirts, gloves, etc., where the employees sit all day, simply feeding the material into machines.

Air currents or drafts are of no material moment in the former shops, while in the latter they will produce great discomfort, if not sickness. Therefore, the latter class should have the main ducts of sufficient area to keep the velocity down to 1,200 to 1,800 ft. per minute and the branches should be proportioned to a velocity of 600 to 1,200 ft.

### COOLING EFFECT OF BLOWER SYSTEM

Another advantage the blower system possesses, infrequently brought to notice, is the cooling and comforting effect it has in oppressively warm weather in the summer time. Simply running the fan will, of itself, greatly relieve the oppressiveness, and when cold water is circulated through the coils the difference is very noticeable.

There is often a wide variation of opinion as to the best temperature to maintain in cold weather. As a general proposition, in foundries and other buildings where the work is active and somewhat vigorous, a temperature of 50° to 55° is sufficient. In machine and wood-working shops, it should be from 60° to 65°. In shoe, clothing and other factories employing help where the work is more or less sedentary, the temperature should be 68° to 72°.

### BASIS FOR DETERMINING AMOUNT OF HEATING SURFACE

The basis for determining the amount of heating surface is the most difficult point to settle. In northern climates, the outside temperature occasionally drops to 10° below zero, and in Minnesota, Dakota and Montana, it goes even lower than 30° below zero. If a plant is designed for these extremely low temperature, it is too large for the conditions prevailing for 95% of the year. It therefore becomes a question of very fine judgment as to just what should be the basis upon which to figure. The following table will give a fair idea of what the resulting temperatures will be under the varying outside atmospheric conditions, the basis being zero.

| CLASS OF BUILDING | Temp. Desired Inside | RESULTING INSIDE TEMPERATURE | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | | Temp. Outside 30° Below 0 | Temp. Outside 20° Below 0 | Temp. Outside 10° Below 0 | Temp. Outside 0 | Temp. Outside 10° Above 0 | Temp. Outside 20° Above 0 | Temp. Outside 30° Above 0 |
| Foundries..... | 50 | 22 | 32 | 41 | 50 | 57 | 63 | 70 |
| Machine Shops..... | 60 | 34 | 43 | 52 | 60 | 67 | 73 | 78 |
| Clothing Shops..... | 70 | 45 | 55 | 63 | 70 | 76 | 81 | 86 |
| Paint Shops..... | 80 | 59 | 67 | 74 | 80 | 85 | 89 | 92 |

From the table it will be noted that for climates where the temperature does not drop below—20°, an inside temperature based on zero outside gives very fair average results; but for climates where the temperature may be from—30° to—10° for several days at a time, the basis is not low enough. For such cases, if the basis is zero, the resulting temperature should be figured about 10° higher than normal, or the temperatures in the table given for machine shops should be taken for foundries and the temperatures for clothing shops should be taken for machine shops.

On the other hand, if the coldest temperature ever reached is zero, and the protracted cold spells are usually at outside temperatures of from 10° to 20° above zero, the basis for determining the size of the plant in zero weather would be 40° at zero for foundries and then taking the tabulated temperatures of foundries for machine shops and the latter for clothing shops.

In this way, considerable unnecessary expense can be saved in the initial cost of the plant; it will be smaller and hence much less costly to operate. It is all well enough to argue that the steam can be cut down on the heating coils and the speed of the fan reduced when the weather conditions moderate, but this is usually not done, simply because it is nobody's business to see that it is done, and instead of thus economizing in the cost of operation, one will invariably find the shop windows and doors open when it becomes too warm inside. Therefore, the more nearly the size of the plant conforms to the needs during average weather condi-

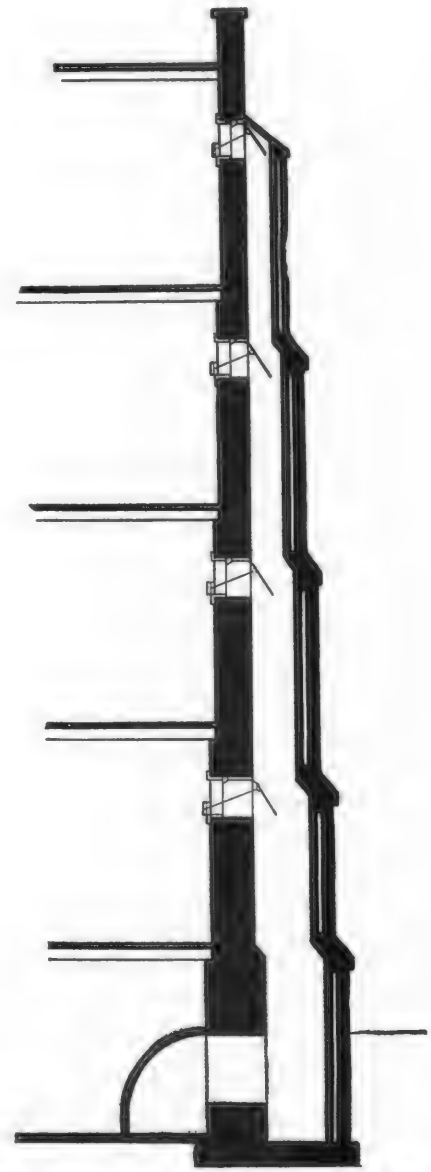


FIG. 5.—METHOD OF HEATING COTTON WOOLEN OR SILK MILLS

tions, the less it will cost to operate it, and this can only be brought about by settling first on what temperature will be satisfactory for the few days of extremely cold weather. This applies to any heating system, and if anything, should be given greater weight for any other than the blower system, as no other can be so readily adjusted to varying weather conditions.

With the blower system one or more sections can be cut out of the heating coils and the blower slowed down in moderate weather; or, steam at a higher pressure can be turned into the coils and the fan be driven at maximum speed for extremely cold weather.

## *Evolution in Heating*

III.

By J. M. W. Kitchen

**The Torridoor Standard Heating System.**

In the December number of this magazine I described a very simple heating system adaptable for installations of very low cost, and yet which would secure to the occupants of the building heated, the necessary qualities of health, comfort and economy, in reasonable efficiency. These three qualities should be present in every heating system, and are practically possible with our present knowledge, heating systems of all degrees of importance.

I now go on to describe what I conceive to be an ideal heating system,* designed to be used in a large range of dwellings occupied by moderate numbers of individuals, and in which a moderate or large sized hot water or steam boiler would be used. In this system is utilized the entirely novel principle of heating the air for respiration with the hot water of condensation, and incidentally cooling the water of condensation to such a low degree, that as feed water, it is enabled to absorb in the boilers, large volumes of relatively low degrees of heat that have hitherto been entirely lost in the chimney gases in steam heating appliances, as well as in steam power generation.

In the past, the water in steam boilers has been maintained at a temperature of over 212° F., and the heating gases passing over the heating surfaces have carried away heat at or above the temperature of the water in the boiler. It is, of course, apparent that unless there is a recipient temperature in the boiler's water is apposition with the heating surfaces, heat in the gases above the temperature of the water connot be reduced to a temperature equal to or below that of the water.

The principle herein applied has grown out of a consideration of the reasons involved for the superior economy of fuel in a hot water heating system. In such a system, especially in connection with greenhouse heating, there is a very large amount of radiation provided, and the water becomes so much cooled before its return to the boiler that it is enabled to pick up lower degrees of heat than would otherwise be the case if the feed water of the boiler was substantially at a temperature of 212° F. or over.

In carrying out this principle we provide a combined air-cooler and water-heater, which is shown in Fig. 9. The water of condensation, and any priming from the boiler, instead of being run directly back to the boiler is conveyed into the top of the air heater and water cooler.

Fresh air for respiration is introduced at the bottom of the device and travels upwardly through it in counter-current to the water, which descends as it is progressively cooled. The water is taken out from the bottom of the cooler and run into the

*The system and boiler design herein shown is controlled by The Torridoor Power, Light and Heat Co. of America.

feed pipe arranged as an inverted syphon to maintain a high water level in the water cooler. The cooled feed water is run into the bottom of the boiler of the system at a low level, and travels through that boiler in counter-current to the travel of the heating gases.

To secure the best results in a system of this kind a boiler should be provided comprising the principles outlined in the structure schematically shown in Fig. 10. In a boiler of this character provision is made for retaining the heat in the fire-box until after combustion is substantially perfected. Over fuel-mass air for combustion is introduced along the front of the boiler, just above the fuel, and only a very shallow length of grate bars is allowed, so that the induced natural draft provided will induce by injection, as it were, a substantial conveyance of the air over the entire front and rear surface of the fuel mass. Provision for extending the grate surface is made by applying sections laterally instead of anteroposteriorly. There is a very large provision made for direct heating surface made at a high level above the fuel mass.

The greatest absorption of heat volumes will be at the level of the crown sheet immediately over the fire. From here the gases travel backwardly and out at a high level gas exit if the fire is being started or forced, or usually, downwardly through the economizer part of the apparatus and out through a low level gas exit.

Feed water is introduced at the lowest level of the economizer section of the boiler, and travels upward and forward in each section, the steam or hot water passing out of the boiler at its highest level.

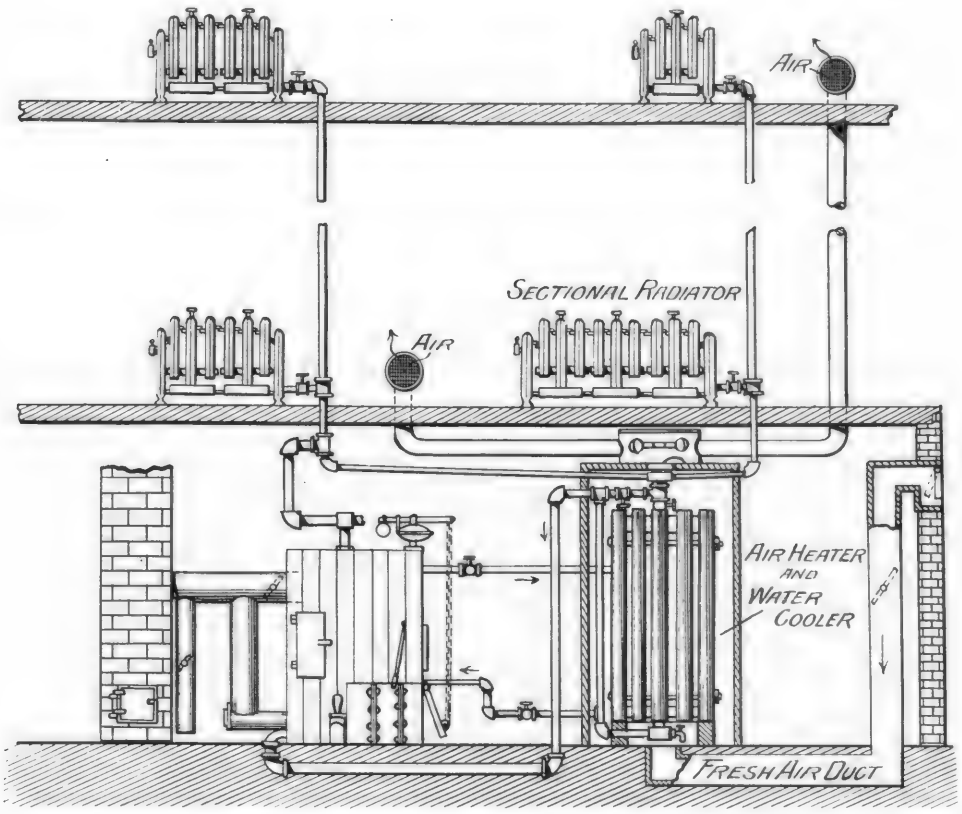



FIG. 9.—ARRANGEMENT OF TORRIDOOR STANDARD HEATING SYSTEM

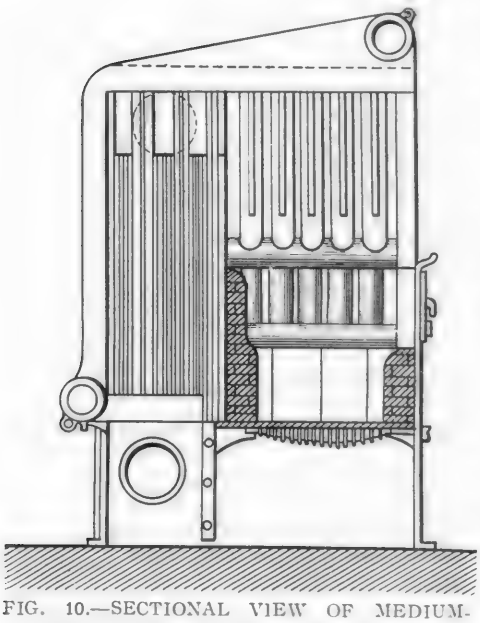


FIG. 10.—SECTIONAL VIEW OF MEDIUM-SIZED TYPE OF TORRIDOOR BOILER

The absorptive abilities of this boiler are highly enhanced by providing fins or processes extending both into the gas passages and into the water space. For boilers of an intermediate size, and requiring great simplicity in management, this design is probably as nearly ideal as can be hoped for. The combustion effected is satisfactorily efficient, though not entirely so.

By studying the arrangement of the system shown in Fig. 9 it will be found that the requirements of health, comfort and economy are all secured through sectional radiation, the introduction of air of the very best quality for respiration, and a hitherto unsupposable economy of fuel is secured. The air passing over heating surfaces of relatively low temperatures will be, of course, of the most desirable quality.

Such a system can be installed at a very slight increase in cost above the simplest old time and unhealthful one pipe steam heating system. The increased cost of installation would probably be returned every year of the use of the system in the fuel economies secured, to say nothing of the comfort and healthfulness of the system.

(To be continued.)

# The Rating of House-Heating Boilers*

By William Kent

## TESTING CAST-IRON HOUSE-HEATING BOILERS

The testing of the evaporating power and the economy of small-sized boilers is more difficult than the testing of large steam boilers for the reason that the small quantity of coal burned in a day makes it impossible to procure a uniform condition of the coal on the grate throughout the test, and large errors are apt to be made in the calculation on account of the difference of condition at the beginning and end of a test. The following is suggested as a method of test which will avoid these errors.

(*a*) Measure the grate surface and weigh out an amount of coal equal to 30, 40, or 50 lbs. per sq. ft. of grate, according to the type A, B, or C, or the ratio of heating to grate surface.

(*b*) Disconnect the steam pipe, so that the steam may be wasted at atmospheric pressure. Fill the boiler with cold water to a marked level, and take the weight of this water and its temperature.

(*c*) Start a brisk fire with plenty of wood, so as to cause the coal to ignite rapidly; feed the coal as needed, and gradually increase the thickness of the bed of coal as it burns brightly on top, getting the fire-pot full as the last of the coal is fired. Then burn away all the coal until it ceases to make steam, when the test may be considered as at an end.

(*d*) Record the temperature of the gases of combustion in the flue every half hour.

*Presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, in New York, January 19-21, 1909.

(*c*) Periodically, as needed, feed cold water, which has been weighed, to bring the water level to the orginal mark. Record the time and the weight.

2. One pound of good anthracite or semi-bituminous coal under the best conditions of air supply, in a boiler properly proportioned, will transmit about 10,000 B. T. U. to the boiler.

CALCULATIONS

Total water fed to the boiler, including original cold water, pounds × (212° — original cold water temperature) = ..... B.T.U.

Water apparently evaporated, pounds × 966 = ..... B.T.U.

Add correction for increased bulk of hot water:

Original water, pounds × $\frac{(62.3-59.8)}{62.3}$ × 966 = ..... B.T.U.

Total ..... B.T.U.

Divide by 966 to obtain equivalent water evaporation from and at 212° F.

Divide by the number of pounds of coal to obtain equivalent water per pound of coal.

The last result may be considerably less than 10 lbs. on account of imperfect combustion at the beginning of the test, excessive air supply, when the coal bed is thin in the latter half of the test, and loss by radiation, but the results will be fairly comparable with other results from boilers of the same size and run under the same conditions. The records of water fed and of temperature of gases should be plotted, with time as the base, for comparison with other tests.

RATING HOUSE-HEATING CAST-IRON STEAM BOILERS

The rating of a steam boiler for house heating may be based upon one or more of several data: 1, square feet of grate surface; 2, square feet of heating surface; 3, coal-burning capacity: 4, steam-making capacity; 5, square feet of steam-radiating surface, including mains, that it will supply. In establishing such a rating the following considerations should be taken into account:

1. One square foot of cast-iron radiator surface will give off about 250 B. T. U. per hour under ordinary conditions of temperature of steam at 212° and temperature of room at 70°.

3. In order to obtain this economical result from the coal the boilers should be driven at a rate not greatly exceeding 2 pounds of water evaporated from and at 212° per sq. ft. of heating surface per hour, corresponding to a heat transmission of 2 × 966 = 1,932, or, say, approximately 2,000 B. T. U. per hour per sq. ft. of heating surface.

4. A satisfactory boiler for furnace for house heating should not require coal to be fed oftener than once in 8 hours; this requires a rate of burning of only 3 to 5 lbs. of coal per sq. ft. of grate per hour.

5. For commercial and constructive reasons, it is not convenient to establish a fixed rate of heating to grate surface for all sizes of boilers. The grate surface is limited by the available area in which it may be placed, but on a given grate more heating surface may be piled in one form of boiler than in another, and in boilers of one general form one boiler may be built higher than another, obtaining thus a greater amount of heating surface.

6. The rate of burning coal and the ratio of heating to grate surface both being variable, the coal-burning rate and the ratio may be so related to each other as to establish condition 3, viz., a rate of evaporation of 2 lbs. of water from and at 212° per sq. ft. of heating surface per hour.

These general considerations lead to the following calculations:

1 lb. of coal, 10,000 B.T.U. utilized in the boiler, will supply 10,000 ÷ 250 = 40 sq. ft. radiating-surface.
1 lb. of coal, 10,000 B.T.U. utilized in the boiler, will require 10,000 ÷ 2,000 = 5 sq. ft. heating-surface
1 sq. ft. of boiler heating-surface will supply 2,000 ÷ 250 or 40 ÷ 5 = 8 sq. ft. radiating surface.

| | Low boiler. | Medium. | High boiler. |
|---|---|---|---|
| 1 sq. ft. of grate-surface should burn... | 3 | 4 | 5 lbs. coal per hour. |
| 1 sq. ft. of grate-surface should develop. | 30,000 | 40,000 | 50,000 B.T.U. per hour. |
| 1 sq. ft. of grate-surface will require.... | 15 | 20 | 25 sq. ft. heating-surface. |
| 1 sq. ft. of grate-surface will supply.... | 120 | 160 | 200 sq. ft. radiating-surface. |
| Type of boiler, depending on ratio heating-surface ÷ grate-surface..... | A. | B. | C. |

TABLE OF RATINGS.

| Type and No. | Square Feet Grate. | Square Feet Heating-Surface. | Coal Burned per Hour. Pounds. | Water Evaporated per Hour. Pounds. | Radiating-Surface Supplied. Square Feet. |
|---|---|---|---|---|---|
| A 1..... | 1 | 15 | 3 | 30 | 120 |
| A 2..... | 2 | 30 | 6 | 60 | 240 |
| A 3..... | 3 | 45 | 9 | 90 | 360 |
| A 4..... | 4 | 60 | 12 | 120 | 480 |
| A 5..... | 5 | 75 | 15 | 150 | 600 |
| B 4..... | 4 | 80 | 16 | 160 | 640 |
| B 5..... | 5 | 100 | 20 | 200 | 800 |
| B 6..... | 6 | 120 | 24 | 240 | 960 |
| B 7..... | 7 | 140 | 28 | 280 | 1,120 |
| B 8..... | 8 | 160 | 32 | 320 | 1,280 |
| C 6..... | 6 | 150 | 30 | 300 | 1,200 |
| C 7..... | 7 | 175 | 35 | 350 | 1,400 |
| C 8..... | 8 | 200 | 40 | 400 | 1,600 |
| C 10..... | 10 | 250 | 50 | 500 | 2,000 |
| C 12..... | 12 | 300 | 60 | 600 | 2,400 |
| C 14..... | 14 | 350 | 70 | 700 | 2,800 |
| C 16..... | 16 | 400 | 80 | 800 | 3,200 |

The table is based on the utilization in the boiler of 10,000 B.T.U. per pound of good coal For poorer coal the same figures will hold good except the pounds coal burned per hour, which should be increased in the ratio of the B.T.U. of the good to that of the poor coal. Thus for coal from which 8,000 B.T.U. can be utilized the coal burned per hour will be 25 per cent. greater.

For comparison with the above table the following figures are taken and calculated from the catalogue of a prominent maker of cast-iron boilers:

| Height. | G Grate. | H Heating-surface. | R Radiating-surface. | $\frac{H}{G}$ | $\frac{R}{G}$ | $\frac{R}{H}$ | B.T.U. per Hour = R × 250 | $\frac{B.T.U.}{H}$ | Coal per hour per sq. ft. grate. * |
|---|---|---|---|---|---|---|---|---|---|
| Low..... | 2.1 | 45 | 210 | 21.5 | 100 | 4.7 | 52,500 | 1,167 | 2.5 |
| | 4.7 | 90 | 600 | 19.1 | 128 | 6.7 | 150,000 | 1,667 | 3.2 |
| Medium.. | 4.2 | 103 | 600 | 24.5 | 143 | 5.8 | 150,000 | 1,456 | 3.6 |
| | 8.2 | 195 | 1,500 | 23.8 | 183 | 7.7 | 375,000 | 1,923 | 4.6 |
| High..... | 6.7 | 210 | 1,200 | 31.3 | 179 | 5.7 | 300,000 | 1,476 | 4.5 |
| | 14.7 | 420 | 3,300 | 28.6 | 225 | 7.9 | 825,000 | 1,964 | 5.6 |

*Equals B.T.U. per hour ÷ 10,000 G.

# Relative Corrosion of Wrought Iron and Steel Pipes

Our readers will remember a paper on the above subject by T. N. Thomson, of Scranton, Pa., which was read at the last annual meeting of the American Society of Heating and Ventilating Engineers and published in THE HEATING AND VENTILATING MAGAZINE for January, 1908.

The interest developed in the discussion of this paper led to the appointment of a committee, composed of T. N. Thomson, chairman; Thomas Barwick and Frank N. Speller, to make further investigations of the relative corrosion of wrought iron and steel pipes. The committee has selected one test in particular which forms its report presented at this year's annual meeting.

The report is prefaced by the statement that the committee is endeavoring to accumulate facts obtained by tests made under working conditions. It is not looking for opinions, because it has reason to believe that very few engineers have had an opportunity to give sufficiently close attention to this subject to correctly identify wrought iron from soft steel pipes while pursuing their regular vocation. This being so, your committee feels that it should make tests itself, or have them done under its own supervision, and it is along this line that we are now endeavoring to collect data.

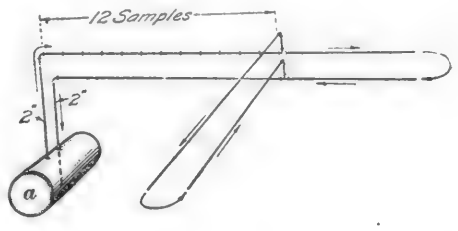



Fig. 2

FIG. 2.—HOT WATER CIRCULATING LINE IN WHICH THE TWO KINDS OF PIPE WERE INSERTED

The committee considers the test a valuable one, inasmuch as it shows the behavior of modern wrought iron and steel pipes when subjected to internal corrosion during actual working service, each sample of pipe in the test being subject to exactly the same working conditions and the same corrosive influences as the others.

The samples were inserted in a hot-water circulating line, as shown in Fig. 2, being located near the ceiling of the engine room of the Instruction Building of the International Correspondence Schools in Scranton, Pa. The samples were inserted November 18, 1907, and were used as part of a hot-water circulating and distributing system that conveys hot water from a boiler *a* to 24 combination faucets at press-room sinks continuously till November 30, 1908, when they were

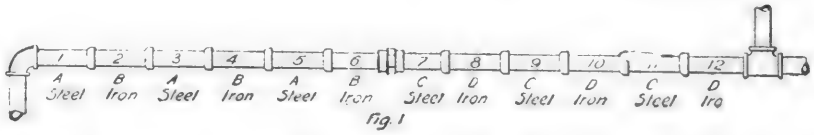



Fig. 1

FIG. 1.—SHORT LENGTHS OF WROUGHT IRON AND STEEL PIPE AS FITTED UP FOR TEST

PIPES USED IN THE TEST

The test was made with six pieces of 2-in. strictly wrought iron pipes and six pieces of 2-in. steel pipes each about 9 7/16 in. long, over all measurements, including the threads, and joined together with wrought iron couplings in the ordinary manner, as shown in Fig. 1. These samples were secured from different sources which we will, for convenience, represent by the letters A, B, C, and D, and were tested out and identified. They were screwed together and inserted on the same line and under exactly the same conditions as existed in the test which was illustrated and described in Mr. Thomson's paper. The test was made, in a measure, as a check on the test previously referred to in that paper.

removed because of a leaky seam which developed in the sample marked No. 8.

The water was heated by a brass steam coil in *a*, and flowed through the samples with a velocity that was never high enough to remove the rust as it formed inside the pipes. The temperature of the water was approximately 200° F., and the pressure varied from about 80 lbs. to 130 lbs. per sq. in.

The accompanying table shows the effect of the corrosion on the different samples.

It will be observed that the steel pipe obtained from A is very much superior to the steel pipe obtained from C. The metal under the deepest pitting in the A samples is .0700 of an inch thick, while the metal under the deepest pitting in the C samples is .0387 of an inch thick. This means that C's pipe will corrode through much more rapidly than A's pipe. The table shows that the weight of each of A's samples was 38½ oz., while C's samples each weighed about 40⅜ oz. for the same length. This means that C's pipe had originally an average thickness greater than that of A's pipe, yet the thicker pipe is considerably less durable than the thinner pipe.

In a similar manner it can be seen that the samples obtained from B are much more durable than the samples obtained from D, even although the original weights and thicknesses were practically the same in both.

It will be seen that the least durable piece of pipe in the test is C's No. 11, which has .0387 inch of thickness under its deepest pitting, while the second and third poorest pieces are D's pipe with a difference of .008 of an inch in favor of the latter.

The samples obtained from C and D can hardly be called first-class pipe when compared with the samples of A and B.

Assuming the samples from A and B to be good quality steel and iron pipe, respectively, their relative lives were then considered. To accurately obtain this it was necessary to find the depth of the deepest pitting and

CORROSION TEST NO. 2

Made in a Hot Water Pipe Line in the International Textbook Company New Printery.

Pipes removed, November 30, 1908.

Pipes put in service, November 18, 1907.

| Boiler End. | 1 Steel. A. | | 2 Iron B. | | 3 Steel. A. | | 4 Iron. B. | | 5 Steel. A. | | 6 Iron B. | | 7 Steel. C. | | 8 Iron. D. | | 9 Steel. C. | | 10 Iron. D. | | 11 Steel. C. | | 12 Iron. D. | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Length in inches end to end of thread | 9 5/16 | | 9 7/16 | | 9 5/16 | | 9 5/16 | | 9 7/16 | | 9 7/16 | | 9 7/16 | | 9 7/16 | | 9 5/8 | | 9 7/16 | | 9 11/16 | | 9 5/16 | |
| Weight in ounces when first put in without couplings | 38 1/2 | | 41 | | 38 1/2 | | 41 | | 38 1/4 | | 41 | | 40 1/8 | | 39 5/8 | | 41 3/8 | | 41 3/8 | | 41 1/4 | | 41 | |
| Weight without couplings in ounces when pipes are taken out and rust removed | 31 5/8 | | 35 1/4 | | 32 1/2 | | 34 1/4 | | 32 1/4 | | 34 1/8 | | 35 1/8 | | 32 1/4 | | 34 1/2 | | 35 1/8 | | 35 1/4 | | 32 | |
| Weight in ounces lost by corrosion | 6 7/8 | | 5 1/4 | | 6 | | 6 1/4 | | 6 1/2 | | 6 5/8 | | 5 1/4 | | 6 7/8 | | 6 7/8 | | 6 1/4 | | 6 1/2 | | 9 | |
| Per cent. of weight lost by corrosion | 17.9— | | 14+ | | 15.6 - | | 15.2 + | | 16.8 - | | 16.2 - | | 13+ | | 17.4 - | | 16.6+ | | 15.1+ | | 15.6 | | 22 | |
| | 1 T. | 1 B. | 2 T. | 2 B. | 3 T. | 3 B. | 4 T. | 4 B. | 5 T. | 5 B. | 6 T. | 6 B. | 7 T. | 7 B. | 8 S. | 8 S. | 9 T. | 9 B. | 10 T. | 10 B | 11 T. | 11 B. | 12 T. | 12 B. |
| Thickness of material under deepest pits | .1264 | .072 | .0750 | .0862 | .0833 | .0906 | .0937 | .0820 | .0935 | .0983 | .1117 | .1225 | .0490 | .0851 | .0683 | .1000 | .0752 | .0540 | .0872 | .0463 | .0814 | .0472 | .0910 | .0632 |
| | .0984 | .0816 | .0842 | .0966 | .0875 | .0901 | .1030 | .0992 | .0935 | .0857 | .1057 | .1150 | .0633 | .0840 | .0827 | .0839 | .0670 | .0638 | .0854 | .0620 | .0516 | .0659 | .0800 | .0460 |
| | .0965 | .0830 | .0830 | .0900 | .0874 | .0940 | .0890 | .1070 | .0935 | .0757 | .0983 | .1055 | .0590 | .0722 | .0598 | .0930 | .0520 | .0620 | .1057 | .0823 | .0406 | .0387 | .0868 | .0578 |
| | .0943 | .0830 | .0957 | .0833 | .0850 | .0887 | .0812 | .1015 | .1000 | .0970 | .1164 | .1047 | .0767 | .0766 | .1078 | .1064 | .0840 | .0640 | .1005 | .0820 | .0547 | .0480 | .0810 | .0480 |
| | .0890 | .0700 | .0930 | .1042 | .0953 | .0841 | .0809 | .0870 | .0944 | .0825 | .1017 | .1160 | .0534 | .0850 | .0789 | .0999 | .0890 | .0695 | .1132 | .0883 | .0668 | .0720 | .0516 | .0720 |
| Least thickness | | .0700 | .0750 | | .0833 | | .0809 | | | .0825 | .0983 | | .0490 | | .0598 | | .0520 | | | .0463 | | .0387 | | .0460 |

compare it with the original thickness of the pipe. Thus:

A's poorest steel sample is No. 1, and the deepest pitting is in the bottom half. B's poorest wrought iron sample is No. 2, and the deepest pitting is in the top half. Fig. 3 shows the depth of the pittings at these points.

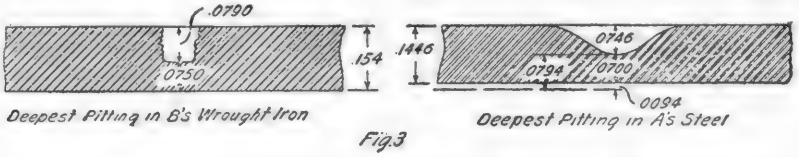



FIG. 3.—DIAGRAM OF THE DEEPEST PITTINGS IN TWO OF THE SAMPLES OF PIPE

The standard thickness of 2-in. full-weight pipe is .154 of an inch, but as the steel samples were 38½ oz. as against 41 oz. for the iron, the original thickness of the steel was practically

$$.154 \times 38.5 \div 41 = .1446 \text{ in.}$$

The exact depth of the pitting in the wrought iron, therefore, is .154 — .0750 = .0790 in., while the depth of the steel is .1446 — .0700 = .0746, showing that the actual depth of the deepest pitting in the poorest piece of A's steel pipe is not as bad as the actual depth of the deepest pit in the poorest piece of B's strictly wrought iron, the difference being .0790 — .0746 = .0044 of an inch in favor of the steel pipe. If we consider the fact that steel is about 2% heavier than wrought iron, the difference will be in favor of the steel.

If A's steel samples had been originally as thick as B's iron samples, the metal under the deepest pitting would have been about .0700 + .0094 = .0794 as against .0750 for the iron.

This test checks up well with Mr. Thomson's paper, and in the opinion of the committee demonstrates that modern steel pipe of good quality is, at least, as durable as modern strictly wrought iron pipe of good quality, and is very much superior to a poor quality of wrought iron in this class of work.

This test suggests that the durability of a pipe depends more upon the quality of the metal than upon its thickness.

The committee calls special attention to the fact that it finds it is not safe to accept reports regarding the corrosion of wrought iron and steel pipes without first identifying the materials, because so many engineers cannot ordinarily distinguish the difference between them.

A few instances are quoted to prove this statement:

At the last annual meeting one of the members brought two 3-in. short nipples and presented them as examples of quick corrosion in steel pipe. They were believed to be steel nipples by some of the members, while others thought they were wrought iron, but nobody had a definite opinion. The committee was then requested to have them taken away and tested, which was done. Each of the nipples was sawed in half and a photograph taken of each of the defective halves. They are shown in Fig. 4. These nipples were tested in three ways by three entirely different people. A test by analysis made by an experienced metallurgical engineer showed the following composition:

| | Sulphur. | Phosphorus. | Manganese. | Carbon |
|---|---|---|---|---|
| B 1 | .022 | .229 | trace | trace |
| B 2 | .026 | .251 | trace | trace |

The analyst's decision is that there is no question of this material being wrought iron.

A fracture test being applied showed without a doubt that the nipples are wrought iron, as can be seen by the broken pieces which were brought to the meeting for inspection.

The acid test was applied by an experienced structural engineer in accordance with the instructions published by pipe manufacturers, and his

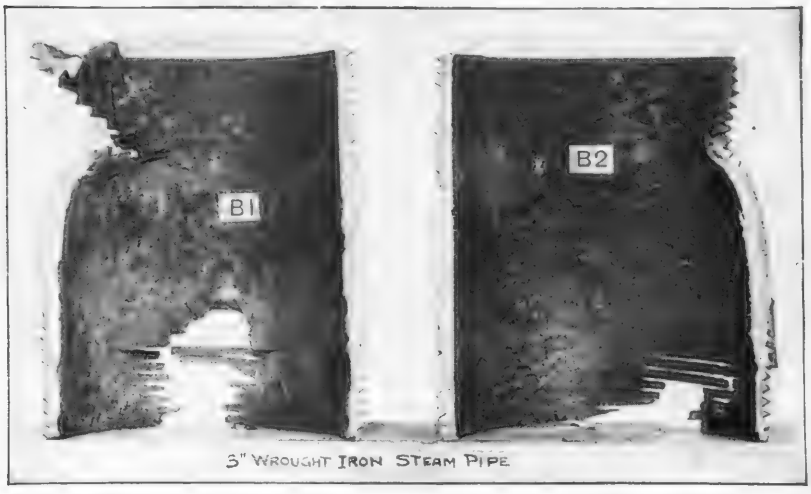



FIG. 4.—CORROSION IN 3-IN. WROUGHT IRON NIPPLES MISTAKEN FOR STEEL

decision is that the nipples are both wrought iron.

So here is a specific case where an expert heating engineer presented to the society two wrought iron samples under the impression that they were steel, and nobody present could prove him wrong.

This example is enough to teach one to be cautious in coming to conclusions regarding the relative corrosion of wrought iron and steel pipes.

## *Air Cooling by Refrigeration**

By W. W. Macon

To obtain the number of tons of refrigerating capacity needed in a refrigerating plant to cool an air supply from any given initial temperature and relative humidity to any desired temperature and relative humidity is the question which this paper plans to discuss and the charts accompanying it are offered as a quick means for getting the result.

Air cooling is seldom merely a case of air warming reversed. In air warming the vapor present is heated along with the air, but it has so little weight in comparison with the air that it is neglected in heating calculations. In air cooling, however, portions of the vapor must be extracted step by step, and this process requires the absorption of the heat of vaporization necessary to reduce the vapor to water.

The heat of vaporization is relatively so large, that the total heat involved in cooling air may be two or three times as much as that involved in warming the same amount of air through an equal range of temperature. In air cooling, therefore, humidity is an important consideration.

Heating 43.35 lbs. of saturated air at ordinary atmospheric pressure from 82° to 83° F. means heating 42.35 lbs. of dry air and 1 lb. of water in the form of vapor. Taking the specific heat of air at 0.238, and assuming that the specific heat of the vapor is the same as for water or unity, the addition of heat involved is $0.238 \times 42.35 + 1 = 11.08$ B. T. U.

Cooling the same weight of saturated air from 82° to 81° means a subtraction of heat of approximately $0.238 \times 42.35 + 0.026 \times 1050 = 10.08 + 27.3 = 37.38$ B. T. U., 0.026

*Read at the annual meeting of The American Society of Heating and Ventilating Engineers in New York, January 19-21, 1909.

being the weight, in a fraction of a pound, of the vapor condensed in the 1° drop, and 1050 being approximately the latent heat of vaporization, or the heat corresponding to that effecting the change of state from liquid to gas or *vice versa*.

It will be noted that over twice as much heat has to be extracted for liquefying the vapor as for cooling the air itself, and this explains the important difference between air cooling and air heating, and why it is so much more expensive to cool air than to warm it. Cooling air without bringing about the saturated condition is, of course, no different than air warming, but an air-cooling problem would always carry the conditions beyond this limitation.

### THE HEAT EXCHANGES IN THE ROOM COOLED

To maintain a room, as in summer, at a temperature below that outdoors means that a flow of heat will occur from the outside to the inside. The supply of heat will be augmented by the people present and by the lights burned. To keep down the temperature is the work of the air supply, which must be in sufficient quantity and sufficiently below the desired room temperature to absorb the heat supply without exceeding the room temperature before escaping.

The refrigerating plant must then be chosen of a size equal to cooling the calculated volume of air from the outside temperature to a temperature low enough to take into account the warming up that the air may get before reaching the room.

Some of the limiting considerations are that the volume of air must not be so great as to cause draftiness, nor must the individual be directly exposed to air currents at low temperature. The lower limit of air temperatures will naturally be dependent in part on the temperatures of the ammonia or brine-cooling coils, and on the depth of the coils in the direction of air flow and on the air spaces between and the velocity of flow through the coils.

Assume that a room is to be at 75° when it is 90° outdoors. The transmission of heat through the exposed walls for the 15° range will be about $\frac{15}{70}$ or $\frac{3}{14}$ of what the transmission is at 70° range, and it might figure that the average flow of heat through the walls is 8 B. T. U. per hour per square foot of the average exposed wall surface. If there is a large area of glass surface including skylights, through which the sun's rays can penetrate, account must also be taken of this source of heat, regarding it, for example, as equivalent to adding 2° to 5° to the outside temperature.

That the animal heat from the individual is something of importance, can be apprehended from the fact that the average individual will give off probably 400 heat units per hour.

If the character of the walls and the relation of the actual glass surface to the total wall surface is such that the average transmission per square foot of the wall surface is about 8 B. T. U. per hour, it will be seen that each individual in the room is equivalent to an additional 50 sq. ft. of exposed wall surface. In other words, if there is a relatively large number of people to be accommodated in the room, the heat supplied from the human beings will be considerably greater than that coming in from the external sources.

Having ascertained the total amount of heat which will be supplied to the room interior under the extreme requirements for which the installation is to be designed, a measure is then had of determining the probable volume of air which should be admitted. This, of course, depends on the number of degrees through which the air is to be allowed to warm from the time it enters the room to the time it passes away.

The determination of the proper number of cubic feet of air in a given time is a matter of arithmetic—the greater the range of temperature the less the volume. If a room is to be maintained at 75° and the air by which this is to be effected is to be

allowed to warm say, 10°, it is obvious that the air must enter the room at a temperature approximating 65°.

As there will be an unavoidable rise in the temperature of the air in its transit from the air-cooling coils and the fan to the room inlets, the actual temperature to which the air must be reduced by the refrigerating installation must be below 65°. Allowing 5° on this account means that the air should be cooled by the refrigerating plant to about 60°.

### DETERMINING THE REQUIREMENTS OF REFRIGERATING APPARATUS

Having ascertained the volume of air necessary by some such process as outlined in the foregoing, it remains to determine how much heat must be abstracted from the air supply to bring about the specified conditions and, therefore, how much refrigerating capacity will have to be provided in the refrigerating plant.

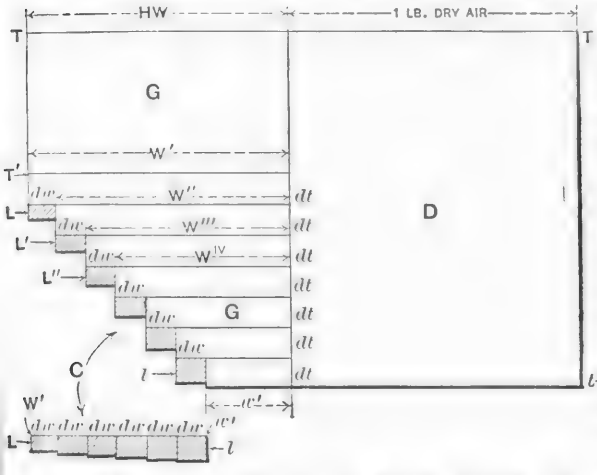



FIG. 1.—DIAGRAM TO SHOW STEPS IN COOLING AIR

Suppose the air received at 90° F. has a relative humidity as high as 80%. Suppose also that it is desired to have the room air at a temperature of 75° and a humidity of about 60%.

Either Chart I or Chart II can be used with these particulars, Chart II being the simpler and therefore more practical, but Chart I affording an opportunity to go into details. The important parts of Chart I are the sections A, B, C and D. Section C gives the number of B. T. U. which must be removed from every pound of air to dispose of the excess moisture; section D gives the number of B. T. U. per pound of air which must be removed to cool the air itself.

### HOW TO USE CHART I

The point B in section A lies on the curve opposite the 75° temperature, which it is desired to have within the space cooled. A line is run vertically downward from the point B until it is opposite the point in section B corresponding to a humidity of 60%. A curved line is then traced parallel with the adjoining curved lines in section B until diagram A is again reached, when a line is run up vertically hitting the curve in diagram A at a point D which is found to correspond to a little above 60°.

The line is then run horizontally into section D, until it intersects the diagonal line corresponding to the initial temperature, in this case, 90°. From the point of intersection running vertically downward in D is obtained a point on scale 1, which shows that 6.9 B. T. U. are required for every pound of air received to cool it through the range of temperature stated.

The dotted lines corresponding to step 2 show that from the point A in the curve of section A opposite the 90° initial air temperature is dropped a line until it terminates in section B opposite the 80% humidity of the air supplied. A curved line is then being traced parallel with the curved lines in B, until section A of the charts is again reached, when a line is run vertically upward to section C, and then run parallel with the diagonal lines of that section, as indicated in dotted lines. From the point D, already determined on the curve in section A, in continued a vertical line upward, as marked, this line continuing into section C until it intersects the diagonal just located in that section. From the point of intersection is run a line horizontally to the scale at the left of section C, which shows the number of heat units required to effect the condensation of the moisture. In this

case it is noted that for every pound of air 14.2 B. T. U. are required.

The total amount of heat, therefore, necessary to cool the air and to remove the moisture is equal to 6.9 + 14.2 = 21.1 heat units per pound of air. The sections of the diagram marked E, F and G can be employed to determine the amount of heat that must be removed to cool the varying quantity of moisture from the initial temperature to the final temperature, but, as will be shown, this amount of heat is relatively small, probably averaging about 2 or 2½% of the amount of heat just determined. Thus for a ready estimation of the amount of heat which must be removed, we may add 2½% to the 21.1 heat units, making a total of 21.6 heat units for every pound of dry air.

From this figure the capacity of the refrigerating plant in terms of so-called tons of refrigeration can, for every 1,000 cu. ft. needed per minute, be obtained by multiplying the number of heat units found by the charts by 0.35.

For example, suppose that 6,600 cu. ft. of air are required per minute. For the case in hand the heat that must be abstracted per pound of dry air is equal to 21.6 B. T. U. Multiplying 21.6 by 0.35 gives 7.56 tons of refrigerating capacity for every 1,000 cu. ft. required per minute. Multiplying this by 6.6 shows that the total refrigerating capacity is 49.9 tons, or, say, 50 tons.

Section A of Chart I shows that to get the air finally at 75° with 60% humidity, the refrigerating plant must be able to cool the air to about 60.5°. This means that after the refrigerating plant has done the work of cooling the air to very nearly 60°, the air can be warmed almost 15°, which fact is merely a coincidence allowing for the 5° rise in the air temperature before the room is reached and the 10° after the air enters the room.

The logical way would have been to take the low temperature at 60°, with the humidity 100%, inasmuch as the air must always leave the refrigerating coils saturated, and then to note if a desired condition of humidity would exist at the room temperature. If too high humidity is likely to result, the air will have to be cooled to a greater extent and the calculations modified accordingly. This point will be referred to in connection with Chart II, but its important bearing is explained in the following:

For example, suppose it is desired to employ a smaller quantity of air than such a calculation as the foregoing indicates as necessary. Then the air will have to be cooled to a lower point. Suppose that the air is cooled by the refrigerating plant to 55°, and assume also that the air as it passes through the ducts is warmed about 5°. Then the air after admission into the room can be warmed 15° before a temperature of 75 is exceeded.

From the charts it will be found that 8.4 B. T. U. must be absorbed for cooling each pound of air from 90° to 55°, and that about 16.4 B. T. U. are required to condense the vapor. This makes a total of 24.8 B. T. U., which, increased by 2½%, for the cooling of the vapor itself, gives a total requirement of 25.4 B. T. U. per pound.

This shows that more refrigeration is needed in this case per pound of air in the ratio of 25.4 to 21.6, but as the air can be warmed 15° after its entrance into the room as against 10° in the case first considered, only two-thirds as much air is needed, and therefore the required capacity of the refrigerating plant is in the ratio of ⅔ of 25.4, of 16.9 to 21.6.

In other words, the refrigerating plant when the air is cooled to 55° is only 78% as large as that cooling the air to 60°, owing to the smaller volume of air sufficing. These figures show the value of moving a minimum volume. There is, of course, a limit depending on the degree of ventilation which the room must have, the minimum amount of air necessary for a proper diffusion of the air and the limiting temperatures which one can obtain in an

economically operating refrigerating plant.

Another point to be mentioned in this case is that if the air is cooled to 55° and then warmed to 75° the final humidity at 75° instead of being at 60% will be about 50%. This would in the average case be cause for satisfaction, the desired point being that room humidity must not exceed 60%.

It is the selection of the range through which the air can be warmed after it is cooled by the refrigerating plant that governs in some measure the relative humidity which the air will have finally in the room. When the air is cooled by the refrigerating plant at 60°, there is present in it all the aqueous vapor that can exist at that temperature, and the weight of vapor remains constant as the air passes over no water, and it has no opportunity for acquiring further moisture. The relative humidity which the air will have will depend on the range of temperature through which it is warmed.

The only factor that affects it slightly is the moisture given to the room air by the individuals in the products of respiration and by gaslights in the combustion of the hydrogen content.

The average supply of moisture by the human body is $\frac{1}{15}$ lb. per hour. The introduction of, say, 2,000 cu. ft. of air per hour per individual from a cooling plant carrying the air to, say, 55°, means an hourly supply of 1.4 lbs. of aqueous vapor in the air as compared with about 0.06 lbs. added from the individual.

### MATHEMATICS OF AIR COOLING

The cooling of air involves the following four steps: 1, the cooling of the air from initial temperature to final temperature; 2, the cooling of the vapor contained in the air until the temperature is reached when the air becomes saturated; 3, removing the heat necessary to bring about the gradual liquefaction of the part of the vapor that must be disposed of; 4, the cooling of the vapor that remains in the air from the temperature when the air first becomes saturated to the final temperature.

The amount of heat that must be removed to cool 1 lb. of dry air or $B_1$ is equal to $0.238 \times (T - t')$, where 0.238 is the specific heat of air, and $T$ and $t'$ are the initial and final temperatures of the air.

The amount of heat removed to cool the vapor until the air becomes saturated is, for every pound of air, $B_2 = H\,W \times (T - T')$, where $H$ represents the percentage of humidity in the air at temperature $T$, $W$ represents the weight, in a fraction of a pound of the vapor present with a pound of air when saturated at temperature $T$; and where $T'$ represents the temperature at which vapor will first begin to condense if the temperature is lowered. It is assumed that the specific heat of the aqueous vapor is equal to that of water or unity.

After the point of saturation of the air is reached, it is obvious that each small drop in temperature means that a small part of the vapor is condensed, leaving the air always in the so-called saturated condition with a gradual condensation of the vapor as the temperature is lowered.

In the diagram, Fig. 1, $H\,W$ represents the actual amount of vapor present with every pound of dry air as the air is received by the refrigerating plant, $W$ being the weight of vapor with one pound of air when saturated at the out-door temperature, and $H$ being the percentage of saturation or the relative humidity. As the temperature is lowered this weight of vapor naturally remains constant (although it will occupy less space, as will also the pound of air at the lower temperatures, due to contraction), until a temperature, designated by $T'$ is reached when $H\,W = W'$ or the weight of vapor which makes 1 lb. of air saturated at temperature $T'$.

For subsequent reduction in temperature let the vertical distance marked $dt$ represent a very small drop in temperature. Then the rectangle, which is equal to the area of

$W' \times dt$, represents the amount of heat abstracted when this weight of vapor is cooled through the very small drop in temperature. When an attempt is made to subject the vapor to a further drop in temperature, a small amount of the vapor designated by $dw$ is condensed, leaving an amount of vapor represented by $W''$. Then the area represented by the product ow $W''$ and $dt$ represents the amount of heat to be taken away in cooling $W''$ lbs. of vapor through a small drop in temperature. On the next small drop in temperature another small amount of vapor is condensed, and the process can be imagined as taking place in successive steps until the final temperature desired is reached, when the amount of vapor existing is represented by the line $w'$. Obviously the area of the long rectangles represents the total amount of heat taken from the vapor.

In a similar way the small shaded rectangles represent the amount of heat abstracted in effecting the condensation of the vapor. If we multiply each small amount of moisture that is condensed by the latent heat of the vapor at that temperature, the product will be the amount of heat that has to be abstracted in each case. As the temperature is reduced the latent heat of vaporization rises slowly, and this is indicated in the shaded sections.

The area of the white rectangles, inasmuch as the little offsets $dw$ are very small, is equal to the area of a trapezoid, of which $W'$ and $w'$ are the bases, and of which the altitude is the sum of all the small drops in temperature, which sum in turn is equal to the difference between $T'$ and $t'$. In other words, the area in the white rectangles, or the amount of heat which must abstract in cooling the gradually decreasing weight of moisture, is equal to

$$(W' + w') \times \tfrac{1}{2} (T' - t') = B_2.$$

If we arrange the black rectangles as shown separately in Fig. 1, it will be seen that their collective area is also that of a trapezoid; that is, that the amount of heat required to condense the moisture is equal to

$$\Sigma\, dw \times \tfrac{1}{2} (L + l) = (W' - w') \times \tfrac{1}{2} (L + l) = B_1.$$

CONSTRUCTION OF CHART 1

Chart I is made to give the results for these various equations. Section A is a curve based on figures given in tables on the properties of air such as are commonly found in books on heating. It shows the relation between the amount of vapor existing with 1 lb. of dry air saturated and the temperature of the air.

Section B is a group of equilateral hyperbolæ, each of which is the locus of points, the products of the coördinates of which is a constant, the coördinates representing the percentages of humidity and the weight of vapor per pound of air. In other words the equation of the curves in section B is $H\,W' =$ constant.

Section C is arranged with diagonal lines, so that by using as indicated by the dotted lines the difference in the weight of vapor in the air at any two points of temperature can be ascertained. Section C is to get the value of the equation $B_1 = (W' - w' + \tfrac{1}{2} (L + l)$.

As the difference between the values of the latent heat is small, particularly over the range which problems of this character cover, a value of 1,070 B. T. U. per pound of vapor was taken as the mean value of $L$ and $l$, and the scale to the left of section C of Chart I is made to read directly in products of 1,070 and the weight of vapor removed from the air.

Section D is also arranged with diagonal lines, so as to give a reading for the difference in initial and final temperature. The scale immediately below D represents the product of this difference in temperature and the specific heat of air, so as to give a reading directly in the number of heat units necessary for cooling every pound of air. This gives the value of $B_1$.

Section G is used to obtain both the amount of heat required to cool the vapor from initial temperature to

the point when the saturation begins, and to obtain the heat required in cooling the vapor after the first point of saturation is reached. The curves of section G are also equilateral hyperbolæ, with differences in temperature as one of the coördinates and the weights of vapor as the other. In determining the amount of heat required to cool the vapor in the first process the figures of the curves must be multiplied by 2. The dotted lines numbered will indicate the application of the curves in section G.

From the point C in the curve of section A, which point corresponds to the temperature when the air becomes saturated, and which point was determined as explained, a line is run horizontally, as marked in the dotted lines, to section F, until the diagonal line corresponding to initial temperature is reached, when the line is dropped vertically downward into section G. This is intersected by a line drawn from section B horizontally from the end of the curve representing the amount of vapor existing with 1 lb. of air at the initial temperature. The point of intersection lies on a curve, the value of which in the present example would seem to be 0.08. Multiply this by 2, and the result will be the number of heat units which must be abstracted per pound of air to cool the vapor in the air from initial temperature to the temperature when saturation occurs. This corresponds to the equation $B_2 = H\,W\,(T - T')$.

The fourth step involves following a horizontal line in section A from the point D, the final temperature to which the air is brought, carrying this line into the section F until it intersects the diagonal line corresponding to the temperature when the air first became saturated. From this intersecting point a line is dropped into section G. This vertical line corresponds to one-half the difference in temperature between the air when first saturated, and when it finally leaves the refrigerating surface, and is part of the equation $B_3 = (W'' + w') \times \frac{1}{2}\,(T' - t')$.

The sum of the vapor contents per pound of air is obtained in the following manner: The line 3 in section E corresponds to the vapor existing in the air as it is received by the refrigerating plant; that is, the amount of vapor existing per pound of outside air. From the point E a line is run at 45° in section E, as indicated, until a vertical line is reached corresponding to the amount of vapor existing at the final temperature, this figure being obtainable from section A in the dotted line marked 1. When this diagonal line in section E reaches this vertical line, a line is run horizontally into section G. Its intersection with the vertical line from section F gives the point desired. This, it will be noted, lies on the curve corresponding to 0.41, meaning that 0.41 heat unit must be abstracted from every pound of air in order to cool the vapor from the temperature, at which saturation occurs to the final temperature, and is the value of $B_3$.

The sum of $B_2$ and $B_3$ shows that 0.57 heat unit should be added to the 21.1 heat units covering $B_1 + B_4$, or that the total heat requirement per pound of air is 21.67. This indicates that the amount of heat shown as necessary by the sections C and D, of Chart I, is increased 2.7% to take into account the cooling of the vapor.

After ascertaining the number of heat units required per pound of air to cool it, it is next desired to know

WEIGHT OF DRY AIR IN 1 CUBIC FOOT OF MIXTURE OF AIR AND VAPOR

| Temp. | Percentage of Humidity | | | | | | |
|---|---|---|---|---|---|---|---|
| Deg. Fahr. | 100 | 90 | 80 | 70 | 60 | 50 | 0 |
| 102 | 0.0659 | 0.0664 | 0.0669 | 0.0674 | 0.0678 | 0.0683 | 0.0707 |
| 92 | .0684 | .0687 | .0691 | .0694 | .0698 | .0702 | .0720 |
| 82 | .0706 | .0708 | .0711 | .0714 | .0717 | .0720 | .0733 |
| 72 | .0727 | .0729 | .0731 | .0733 | .0735 | .0737 | .0747 |
| 62 | .0747 | .0748 | .0750 | .0751 | .0753 | .0754 | .0761 |
| 52 | .0766 | .0767 | .0768 | .0769 | .0770 | .0771 | .0776 |
| 42 | .0784 | .0785 | .0785 | .0786 | .0787 | .0788 | .0791 |

the refrigerating capacity. As the figures are obtained per pound of air, and as the weight of dry air in 1 cu.ft. of a mixture of air and vapor depends not only on the temperature of the air, but on the relative humidity, it would seem necessary to multiply the number of heat units found by the chart by the weight of dry air in 1 cu. ft. of the outside air to ascertain the number of heat units required in the refrigerating plant per cubic foot of the air supplied. It is obvious that the refrigerating capacity should be given in terms of the number of cubic feet of air which

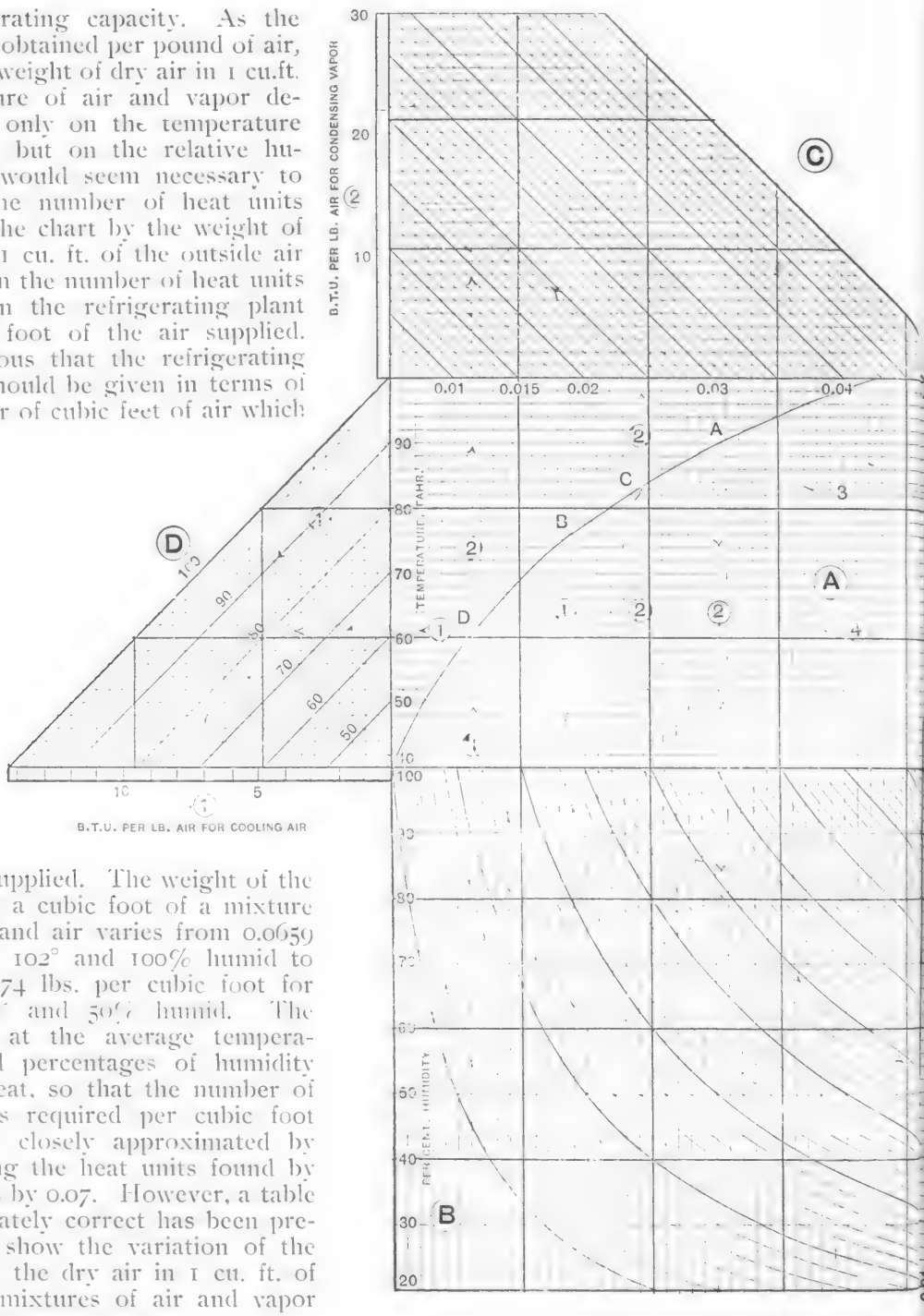


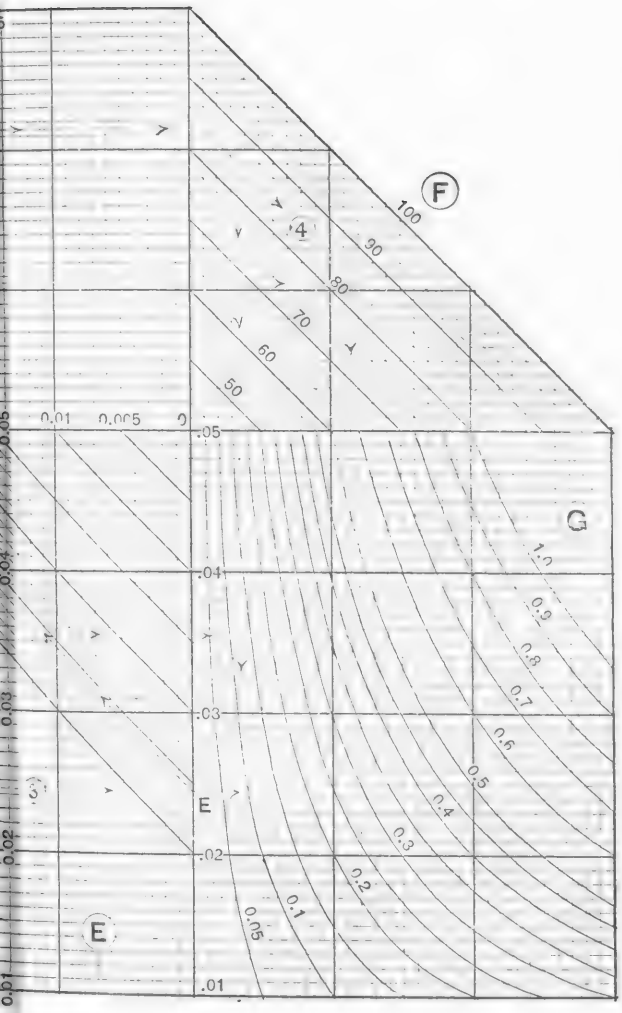

CHART 1. TO OBTAIN REFRIGERATING CAPACITY

must be supplied. The weight of the dry air in a cubic foot of a mixture of vapor and air varies from 0.0659 for air at 102° and 100% humid to about 0.074 lbs. per cubic foot for air at 72° and 50% humid. The variation at the average temperatures and percentages of humidity is not great, so that the number of heat units required per cubic foot would be closely approximated by multiplying the heat units found by the charts by 0.07. However, a table approximately correct has been prepared to show the variation of the weight of the dry air in 1 cu. ft. of different mixtures of air and vapor at different temperatures.

If we multiply the total volume of air required, say, in a minute's time, by the weight of this air in pounds per cubic foot, as shown in the table, and multiply this in turn by the number of B. T. U. found necessary per pound by the charts, the final product will give the total number of heat units which must be removed in the given time. If we divide this by the heat equivalent of 1 ton refrigera-

tion the quotient will give the refrigerating capacity.

One ton refrigeration may be defined as being equal to the heat transfer involved in the freezing or melting of 2,000 lbs. of ice in 24 hours. The latent heat of ice, that is, the number of heat units which must be abstracted from water after it has reached the point of freezing before ice is formed, is equal to 142.2 B. T. U. For 2,000 lbs. this repre-




BRITISH THERMAL UNITS PER POUND OF AIR

sents 284,400 B. T. U. per day, or about 12,000 B. T. U. per hour, or 200 B. T. U. per minute. In other words, 1 ton refrigerating capacity for one day is equivalent to the performance of 200 B. T. U. every minute.

If we divide the total heat requirements as calculated by 200, the result will be as stated, the refrigerating capacity required. If we take the average weight of the dry air in a cubic foot of the average mixture of vapor and air out-doors in summer as 0.07 lbs., every 1,000 cu. ft. of air handled per minute represents 70 lbs. of air to be handled by the refrigerating plant every minute. If one heat unit of refrigeration is required for every pound of air admitted to the system, then 70 heat units will be required per minute per 1,000 cu. ft. per minute. Dividing the 70 heat units by 200 heat units corresponding to 1 ton refrigeration gives a quotient of 0.35 as the number which multiplied by the heat requirements of 1 lb. of air gives the number of tons refrigeration per 1,000 cu. ft. of air per minute.

### APPLICATION TO NEW YORK STOCK EXCHANGE

A practical application of the charts may be had in connection with the refrigerating plant which has been installed in the New York Stock Exchange. This equipment is one of the particularly interesting features of the mechanical plant designed for that building by Alfred R. Wolff of New York City. The plant is notable not alone for its magnitude, although of no less than 450 tons' refrigerating capacity in three units, but for the fact that it represents a solution of the problem of cooling air for ventilation on a scale not considered up to that time.

The apparatus is provided to secure a room temperature of not more than 75° and a humidity of about 55% when air received at 85° F. and 85% humidity. In Chart II we locate two points, one on a line opposite 85° and midway between the curved lines corresponding to 80% and 90% humidity, and the other on a line opposite 75° and midway between the curved lines corresponding to 50% and 60% humidity. The first point corresponds

to the condition of the outside atmosphere, and the second to the condition which it is desired to have inside. From the latter point is dropped a line until the curve corresponding to 100% humidity is reached. This point will be found to lie opposite 58°, and indicates the temperature to which the air must cer-

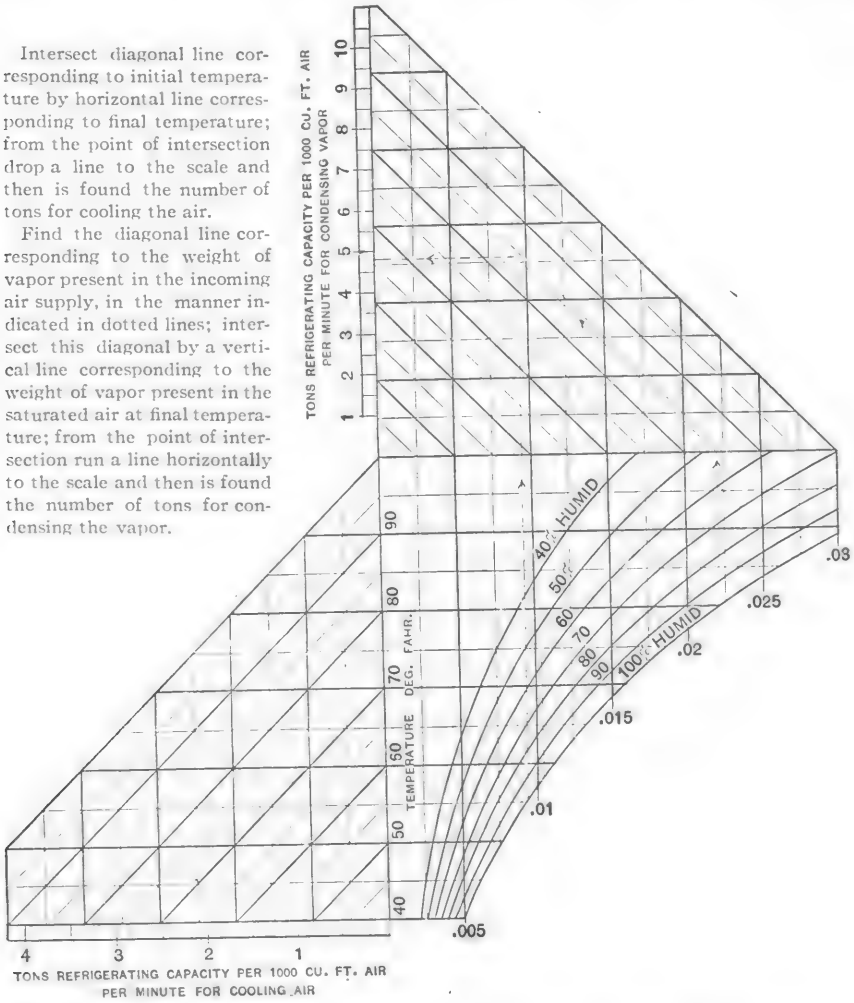

Intersect diagonal line corresponding to initial temperature by horizontal line corresponding to final temperature; from the point of intersection drop a line to the scale and then is found the number of tons for cooling the air.
Find the diagonal line corresponding to the weight of vapor present in the incoming air supply, in the manner indicated in dotted lines; intersect this diagonal by a vertical line corresponding to the weight of vapor present in the saturated air at final temperature; from the point of intersection run a line horizontally to the scale and then is found the number of tons for condensing the vapor.
TONS REFRIGERATING CAPACITY PER 1000 CU. FT. AIR PER MINUTE FOR CONDENSING VAPOR
1 2 3 4 5 6 7 8 9 10
TEMPERATURE DEG. FAHR.
90 80 70 60 50 40
40% HUMID
50%
60
70
80
90
100% HUMID
.03
.025
.02
.015
.01
.005
4 3 2 1
TONS REFRIGERATING CAPACITY PER 1000 CU. FT. AIR PER MINUTE FOR COOLING AIR

CHART II.—TO OBTAIN REFRIGERATING CAPACITY DIRECTLY IN TONS

tainly be reduced to bring about the desired conditions. Running horizontally from this point into the group of diagonal lines until the diagonal line corresponding to the initial temperature of 85° is reached gives a point, which by dropping a line to the scale shows that 2.3 tons' refrigerating capacity are required in the refrigerating equipment for cooling the air for every 1,000 cu. ft. which may be required per minute.

To determine the number of tons necessary for condensing the vapor a line is carried vertically upward from the point corresponding to outside air conditions until the upper group of diagonal lines is reached, when this line is then carried parallel with the diagonal lines. From the point corresponding to inside conditions a line is carried up vertically until it reaches the diagonal line just located, and the point of intersection will be found to lie opposite a position on the scale corresponding to 4.35 tons' refrigerating capacity. The total is

thus 6.65 tons, which should be increased 2½% to take care of the cooling of the vapor, making a total of 6.82 tons for every 1,000 cu. ft. required per minute.

It is stated that the hourly amount of heat added to the board room of the building, comprehending the heat transmission from outside atmosphere through the windows, doors, outside walls and skylights, and taking into account also the heat from the sun, the animal heat from individuals within the room and the heat from lights which may be burning, is altogether about 666,000 B. T. U. If we assume that the air after it leaves the refrigerating plant is warmed 5° before it enters the room, it reaches the board room at a temperature of 63°. As the air is not to exceed 75° within the board room, the air supply which is to offset the heat in the room can be warmed through $75 - 63 = 12°$.

One cubic foot of air heated 12° takes up $1 \times 12 \times 0.075 \times 0.238 = 0.214$ B. T. U. The 666,000 B. T. U. supplied in an hour is equal to 11,100 B. T. U. in a minute, therefore the air supply needed in a minute is equal to $11,100 \div 0.214 =$ about 51,800 cu. ft. per minute. The refrigerating capacity under these conditions would, therefore, be equal to $51.8 \times 6.82 = 353$ tons' refrigerating capacity. The relative humidity, however, will really be more than 55%, inasmuch as the individuals are moisture producers, and if there are gaslights an additional amount of moisture will be added from this source. We may, therefore, see what the effect will be of reducing the air to a temperature of 55° in the refrigerating plant.

By referring to Chart II it will be found by running horizontally along the line corresponding to 55° until it intersects the diagonal line corresponding to 85° that 2.55 tons are required for cooling 1,000 cu. ft. of air per minute. Similarly a line is run vertically upward from the point on the 100% humidity curve corresponding to 55°, until this vertical line intersects the diagonal line corresponding to the 85°, 85% humidity point. The point of intersection shows that 4.8 tons' refrigeration are required for condensing the vapor. The total, adding 2½% for cooling the vapor, is 7.53 tons per thousand cubic feet of air per minute.

In this case we may assume the air is introduced into the board room at 60°, with the opportunity of warming 15° further during the process of supplying the heat to the board room. Warming each cubic foot of air 15° is then equal approximately to 0.268 B. T. U. Therefore the number of cubic feet of air required in a minute is equal to $11,100 \div 0.268$ or about 41,400 cu. ft. Therefore the size of the refrigerating plant necessary is equal to $41.4 \times 7.53 = 312$ tons. There were provided for the board room two 150-ton refrigerating plants, or 300 tons altogether.

Pipe coil surface is employed for cooling the air, and for the board room just described there are 20,000 lin. ft. of 1¼ in. galvanized pipe. This bunker of pipe coil surface is arranged in 22 coils, each coil lying horizontally with the supply and return ends of the 22 coils connected into two vertical headers. The coils are of continuous piping from supply to return header bent in serpentine form, which brings the pipe back and forth across the bunker chamber. The entire bunker occupies a space of about 10 ft. in depth in the direction of air flow and 18 ft. transversely with respect to the air flow.

The connections of the coils to the header are spaced about 4½ in. on centers, and the paralleling or serpentine form of the pipes of each coil brings the piping in two layers or levels, so that the pipes are roughly center to center measurement from the bottom to the top of the bunker chamber about 2½ in. apart, with the pipes disposed in 44 layers. There is a free cross section for the flow of air of perhaps 45 or 50 sq. ft., and with the delivery by the fans of 41,400 cu. ft. per minute the velocity through the bunker chamber ap-

proximates on this basis 900 to 950 ft. per minute.

The 20,000 lin. ft. in the bunker for the board room represents 8,700 sq. ft. of heat-absorbing surface. The 312 tons of refrigerating capacity just figured means that the bunker is to absorb in one hour's time 312 × 12,000 = 3,744,000 B. T. U. Dividing this by the number of square feet indicates that each square foot of the bunker surface is depended on to absorb about 430 B. T. U. per hour.

With air received at 85° and discharged at 55°, the average temperature of the air is 70°. If the brine circulating through the coils is delivered at zero, and leaves the coils at 10° F., the mean temperature of the brine would be 5°. The difference in temperature between the mean temperature of the brine and the mean temperature of the air is thus 65°. This would, therefore, mean that each square foot of the bunker surface is calculated to absorb somewhere in the neighborhood of 7 B. T. U. per square foot of bunker surface per hour per degree difference in temperature between the air and the cooling medium. If the brine should average as high as, say, 20°, the difference between the mean air temperature and mean brine temperature would be 50°, and the amount of heat absorbed by each square foot of the bunker surface per hour per degree difference would approximate 8 B. T. U. It appears also that the refrigerating machinery in the Stock Exchange, which is of the ammonia-absorption type, is operated entirely with exhaust steam from the mechanical plant at not over 4 lbs. pressure above the temperature, and that the water used in the apparatus is used in part for boiler feed purposes and in part for supplying the plumbing fixtures of the building.

### Death of Alfred R. Wolff

Alfred R. Wolff, one of the best-known heating and ventilating engineers in the country, died quite unexpectedly January 6 at his home, 15 West 89th street, New York. He had attended a concert during the evening with his wife and was feeling in good spirits when he returned to his home. He died shortly after, as he was about to retire for the night. He is survided by a widow and four children.

Mr. Wolff had the reputation of having constantly on the boards more work than any other heating and ventilating engineer in the country, the plans frequently representing buildings aggregating in value millions of dollars.

Mr. Wolff was born in Hoboken, N. J., March 15, 1859, receiving his entire school education in that place, and was graduated from Stevens Institute of Technology in 1876. Soon after graduation he became assistant to Chas. E. Emery, C. E., whom he left in 1879 to enter the United States Revenue Marine service as assistant engineer.

He was a charter member of the American Society of Mechanical Engineers.

In 1880 he started a general engineering practice, locating at 38 Park Row, moving to 130 Fulton street in 1892.

He took up the design of steam boiler, heating and ventilating plants, and rapidly acquired a very extensive practice. This practice extended over a period of 28 years. He was the author of "Ventilation of Buildings" and "The Heating of Large Buildings."

Early in life he became associated with Dr. Felix Adler, and devoted much time to the interests of the Ethical Culture Society, first as secretary of the executive committee, and later as trustee and chairman of the board of governors.

He was a member of the Engineers' Club and an alumni trustee of Stevens Institute of Technology from 1893 to 1896, and was permanent trustee from 1900 to the time of his death.

He was engaged in numerous benevolent enterprises, being vice-president of the Charity Organization Society and chairman of the advisory committee of the Baron De Hirsch Trade School.

### Two-Cent Postage in Germany

The two-cent letter postage rate, arranged between Germany and the United States, went into effect January 1, 1909. This applies, however, to letters carried by direct mail lines between the two countries. Letters sent to Great Britain with a two-cent stamp affixed will not be forwarded to Germany unless additional postage is paid.

WE HAVE nothing but admiration for a journal that can obtain credit for matter reprinted in the proceedings of an engineering society, notwithstanding such a society's announced policy and practice not to extend such credit.

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TWO MOVEMENTS of much significance in the engineering world are up for discussion with chances more than even for a successful outcome. One of these has as its object the universal adoption of the Briggs standard for screw threads. The American Society of Mechanical Engineers, and the American Gas Institute have been for some time quietly at work on the project, the mechanical engineers being represented by a committee composed of Edwin M. Herr, first vice-president of the Westinghouse Electric & Mfg. Co., chairman; Wiliam J. Baldwin, New York; George M. Bond, Hartford, Conn.; and Stanley G. Flagg, Jr., Philadelphia.

Both of these organizations are represented at an international conference on the subject being held this month in Paris, and there is apparently much reason for hope that the argument submitted by the American engineers will carry the day.

THE OTHER movement is one that will enlist the interest of every heating engineer who follows its progress. It is nothing less than a concerted and, we think, serious effort on the part of all concerned to reform the ratings of house heating boilers. It is possible the scope of the movement will be enlarged to cover power boilers. Never before, certainly not in recent years, have so many interests undertaken to go into this matter with such carefully laid plans, or with such definite ideas of what they want to have done. Before these words are read a conference in which will be represented The American Society of Heating and Ventilating Engineers, the Boiler and Radiator Manufacturers' Association and the National Association of Master Steam and Hot Water Fitters will have been held in New York at the headquarters of the heating engineers' society for the purpose of taking up the ratings of house heating boilers. The executive committee of the Boiler and Radiator Manufacturers' Association is composed of P. H. Seward, New Yory; R. D. Reed, Westfield, Mass.; Henry Richardson, New York; Harry S. Martin, Dunkirk, N. Y.; and Charles K. Foster, Chicago. The master steam fitters are represented by E. M. Nichols, Philadelphia, chairman; Burton P. Hall, New York, and T. B. Cryer, Newark, N. J. Although the heating engineers have a committee on the subject composed of J. H. Kinealy, chairman; J. R. Allen, A. B. Franklin, W. M. Mackay and B. S. Harrison, the subject is before the society as a committee of the whole during the period of the annual meeting.

# A Notable Differential Thermostat for Humidity Control

Many attempts have been made in the past to devise some device that was quickly and positively susceptible to changes in the relative humidity of the air and that could be used as a controlling device to regulate a humidifying system. Nearly all these inventions have depended for their action upon the hygroscopic properties of silk, hair, wood or other animal or vegetable substance. Such devices have never proved successful for either indicating or controlling humidity, chiefly on account of the time element necessary for the material to absorb or give off moisture thus giving a lagging effect which is not usually quick enough to give practical and accurate control. Then the hygroscopic properties of a material do not remain constant, as the power of absorbing or giving off moisture diminishes gradually although very slowly in some substances.

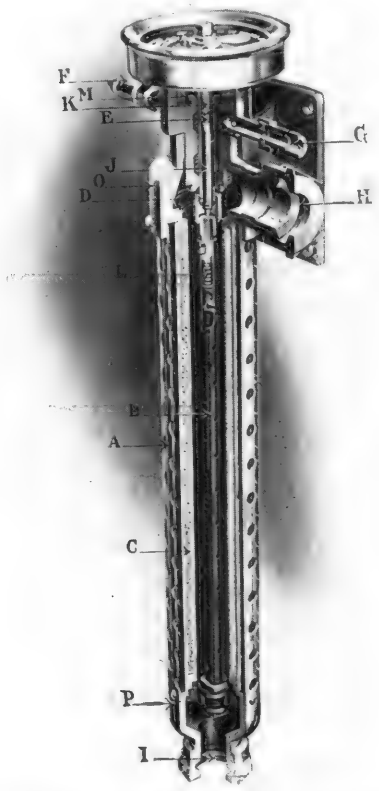



FIG. 1. — NEW DIFFERENTIAL THERMOSTAT FOR HUMIDITY CONTROL

Another method, in an endeavor to accomplish the control of relative humidity, employs a wet and dry bulb thermometer, through which an electric current is passed. The variation in resistance of the electric current due to the height of the mercury columns is made use of to control the humidifying system.

While it is probable that such an arrangement could be adjusted to give a fairly close control that would operate within a certain range in temperature, still such a device is too complicated, too susceptible to the many ills of an electric current, requiring the adjustments to be made by an expert, and so is not at all suitable for practical use.

Mr. W. H. Carrier, consulting engineer for the Carrier Air Conditioning Company of America, 39 Cortlandt street, New York, began work on this problem several years ago along entirely original lines. He developed several schemes, all of which, excepting his differential thermostat, were abandoned because they did not fulfill the requirements which he considered necessary for such an instrument. He wanted an instrument that would respond quickly to any changes in the hygrometric conditions that was simple yet positive in action and strong in construction, and easily adjusted for different percentages of humidity, and above all, he wanted an instrument that was accurate at all temperatures between 65°F. and 105°F. He succeeded in combining all these in a device known as a differential thermostat, illustrated in Fig. 1.

The differential thermostat, although invented in 1906, was not put on the market until the past year, as it was thought best to thoroughly test it practically before offering it for sale.

The instrument when used in connection with the Carrier system of air washing and humidifying has one expanding element, B, subjected to the dry bulb temperature, and another expanding element, A, subjected to the dew point temperature of the room and acting in opposite directions, thus taking advantage of the fact that for a given percentage of humidity the dew point temperature is approximately a fixed number of degrees lower than the dry bulb temperature.

The number of degrees the dew point is below the room temperature for several percentages of humidity is as follows:

| | | |
|---|---|---|
| 85% | approximately..... | 6° |
| 80% | " ..... | 7½° |
| 75% | " ..... | 9° |
| 70% | " ..... | 11¼° |
| 65% | " ..... | 13¼° |
| 60% | " ..... | 16¼° |

It will be noted by referring to the illustration that the two expanding elements, A and B, are made of metal tubing perforated to allow the air to freely circulate on each side, thus quickly imparting its temperature to the metal.

The dew point member A is rigidly fastened to the head O and is free to expand in the direction of I.

The dry bulb member B is rigidly fastened to and supported by the dew point member A, so the dry bulb member B is free to expand in the opposite direction to the dew point member A.

Between the two members is a double tubing and mineral wool insulation to prevent one temperature from affecting the other.

In the head is carried a small double seated compressed air valve J, having seats at E and D. The valve is kept on its seat at D by a small spring K, unless it is held off this seat by the expansion of the dry bulb member B or by the contraction of the dew point member A.

FIG. 2.—LEVER DIAPHRAGM OPERATED BY DIFFERENTIAL THERMOSTAT FOR CONTROLLING AIR DAMPERS

From the above it is seen that the instrument is very simple, really consisting of two members expanding in opposite directions, and one double seated air valve so arranged that a *variation in the difference in temperature* immediately affects the position of the air valve.

In operation all except the head of the instrument is inserted in the air duct leading from the humidifier, and as the air is always saturated in the humidifier, it is at the dew point temperature, which immediately imparts its temperature to the dew point member A. A pipe connection heavily insulated is made from the union H to room in which the humidity is to be controlled, and a similar connection is made from I to the inlet of the ventilating fan, thus what might be termed as a sample of air at the room temperature is passed over the dry bulb member B.

A diaphragm controlled valve is placed on the steam line used for heating the spray water, and a lever diaphragm is connected to a damper which controls the relative amounts of air taken from the outside or returned from the building by the humidifying apparatus.

A small amount of compressed air enters the instrument at F, passes into the chamber M through the air valve E out the pipe G to the diaphragm controlled valve and the lever diaphragm.

If the percentage of humidity is too low the depression of the dew point temperature below the dry bulb temperature will be too great, thus contracting member A and expanding member B, so that the end of member B will be brought against the air valve J, closing E and opening D, thus reducing the air pressure on the diaphragm and allowing more steam to enter the spray water or more warm air to be returned from the building, thus increasing the percentage of humidity.

If the humidity becomes too high, the reverse action takes place.

The pointer on the dial indicates the percentage of humidity for which the instrument is set. By means of a key, which fits the square nut in the center of the dial, the instrument is easily and instantly adjusted to control the humidity at any point.

The spring L is to prevent the member B from jambing the valve too hard on its seat D, thus making it in a sense fool proof.

One of these instruments installed has not been adjusted in any manner in eighteen months, and is controlling the humidity within 2% continuously.

The differential thermostat is now in use in a number of buildings and the makers have yet to receive a complaint regarding their operation.

## New Book

**Tables for Calculating Sizes of Steam Pipes for Low Pressure Heating** is a new book by Isaac Chaimovitsch, just published by Domestic Engineering, of Chicago. The author's methods of calculation are illustrated by a number of typical layouts. The volume includes four tables for calculating steam pipe sizes with boiler pressures at 3, 5 and 10 lbs., respectively, and for various sizes of pipe running from ¾ in. in diameter up to 24 in. Size, 4½ in. x 7 in. 48 pp. and four inserts. Price, $2.00.

## Back Numbers Wanted of The Heating and Ventilating Magazine

Fifty cents apiece will be paid by the undersigned for the following-named back numbers of THE HEATING AND VENTILATING MAGAZINE:

February, 1905.
March, 1905.
November, 1905.
March, 1906.

Address The Heating & Ventilating Magazine Co., 1123 Broadway, New York.

## A New Type of Air Heater for Use with Live or Exhaust Steam, Hot or Cold Water, or Brine

Air and water within the pipes are the bane of most steam air heaters coils and radiators. It will be found that in many of them the steam is not forced to circulate positively through the coils, but can pass directly within the "header" casting from the inlet to the outlet without going through the pipes at all, and, contrary to the usual assumption, this very thing may happen if there is air in the pipes. The loops of pipe employed in such heaters, in fact, make nice air pockets. This possibility is obviated in an interesting manner in the new type of the Green patent air heater.

From an examination of the accompanying diagram showing the course of circulation in this type of steam air heater, it will be seen that the current of steam must follow straight ahead through half the tubes, then back through the other half down to the bottom outlet, driving before it any air that may be present. This is the natural direction of flow, since the air is heavier than the steam. On the other hand, in a steam heater section with a single header at the bottom, the air is expected to flow down through small pipes against the rising current of steam, and under such circumstances expulsion of the air either will not take place at all, or, at the best, very slowly.

Unless provided with a double set of steam traps for the removal of condensation simultaneously from both chambers, the inlet and outlet chambers of all these bottom header heaters must be inter-connected by a direct passage for drainage, so there can be no difference in steam pressure between the two ends of the loops of pipe and the only force tending to remove the air is gravitation.

In the Green patent heater the pressure of the steam is employed directly and positively to push the air through and out of the pipes. With this heater, the preliminary "blowing out" sometimes practiced for the removal of air is really effective. The heater cannot become air-bound, and all of its surface is always active in transmitting heat. While all heaters show a practically equal efficiency in heat transmission per square foot of surface under equal conditions when they are properly cleared of air, the fact is that the construction of many of them is such that they will not clear of air under operating conditions and will not always operate satisfactorily, often failing in the coldest weather when the need for heat is most urgent. Under such conditions, "single

NEW TYPE OF HOT BLAST HEATER, WITH ONE OF THE HEADERS REMOVED, SHOWING ENDS OF TUBES AND DIVISION PLATE

header" heaters frequently allow steam to blow directly from the inlet chamber to the outlet chamber and to escape to atmosphere, although at the same time a large part of the heater is air-bound and cold. The Green heater is so constructed that every pound of steam will be condensed, up to the full theoretical

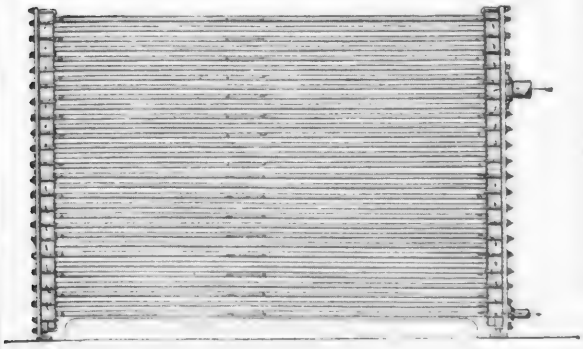


DIAGRAM SHOWING POSITIVE CIRCULATION OF STEAM THROUGH NEW GREEN HEATER

capacity of the heater, and no steam will escape through the heater until every part of its is steam hot.

Practical steam heating men well know that it is always a bad plan to figure on having steam and water flow in opposite directions in the same pipe, since water hammer is apt to occur. Nevertheless, many heating coils are so constructed that the water must come down through the same pipes that carry the steam up. In the Green patent heater, currents of steam and water are always in the same direction and, should water of condensation accumulate, it would be pushed through positively in the same manner that the air is pushed through, as above described.

### CIRCULATION OF HOT OF COLD WATER OR BRINE

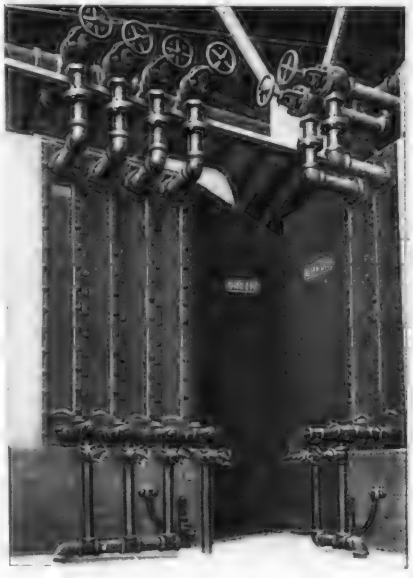
This positive circulation principle has a further advantage. It enables the heater to be used as a hot water heater or as an air cooler, since hot water, cold water or cold brine can be circulated continuously through the pipes and with small expenditure of power. The inlet and outlet openings are so placed that the same head or pressure acts upon each tube, insuring an even distribution of the circulating fluid.

### AS AN AIR DRYER

By arranging the sections of the Green heater in two groups and circulating cold water or brine through the one group and warm water or steam through the other, the heater becomes an effective air dryer, possessing an advantage in that the cooling process does not introduce into the air a mist of water particles which can afterwards be removed only imperfectly by means of baffles and mechanical separators.

Ease in caring for a heater is next in desirability to freedom from interruption and from trouble in operation. Exhaust steam usually carries with it grease, oil, etc., which in time so coat the inner surfaces of steam coils as to interfere seriously with the transmission of heat. A coil made up of bent pipe or pieces of pipe fastened together by ells is hard to clean, the more so, if the only way to get at the tubes is by unscrewing them from their sockets, as with heaters made with solid headers. In such cases it is, in fact, practically impossible to remove oil and grease from the tubes by mechanical means and about the best that can be done is to scour it out with a solution of caustic soda or lye.

The same considerations have weight with heaters or coils in which hot or cold water or brine are circulated. Hot water, especially, may deposit scale, which it frequently does in closed feed water heaters and steam boilers, and about the only practical way to remove this scale is to bore or scrape it out with tube cleaners. This is impossible



NEW GREEN HOT BLAST HEATER VALVED FOR LIVE AND EXHAUST STEAM

in a crooked pipe connected with ells, as will be seen by referring to the outline drawing of the miter-coil heaters The cleaning of the Green patent heater, whether of oil, grease or scale, is strikingly simple. After removing one or both of the headers the tubes can be swabbed or scraped out at once and by holding a candle at one end and look-

ing through from the other, their condition is easily seen.

The headers of the Green heater stand in a vertical position at the sides of the heater casing and when the covers are removed all the tubes are exposed in convenient position for inspection or cleaning or for removal or repair. The tubes are rolled into the headers like boiler tubes and can be removed or replaced by anyone who has skill enough to replace a tube in a boiler. If it is not desired to put a heater out of service for tube repairs, steam can be shut off from the injured section while the two ends of the defective tube are plugged. The screwed joints generally used frequently leak, and in order to reach the defective joint it may be necessary to dismantle a third of the heater, thus running a chance of making many leaky joints while attempting to remedy one. Under such circumstances leaks are often allowed to go on indefinitely. In the Green heater each pipe is individually accessible and repairs are easily made, the number of joints is also reduced fully one-half.

The cover plates on the tube headers of the Green patent air hearter are both easily removed and easily replaced. They are held in position by a row of hood bolts, the threaded ends of the bolts projecting through holes in the header cover plate. This construction is more practical than the use of studs, which are liable to be bent or injured by tools in removing or replacing the cover plate and which, when once so injured, can be extracted and replaced only with difficulty. The hook bolts used on the Green heater are not liable to injury and can be replaced without effort.

In this new construction, as shown on the accompanying cross-sectional view, the making of a joint has been simplified by substituting a rope for the usual gasket. This rope can be laid roughly in place in the groove without fitting, cutting or cleaning, and when the cover is drawn down tightly against it by means of the heavy hook bolts, it is absolutely steam tight against the highest steam pressure. This rope can be obtained at less cost than a good grade of gasket.

The advantages described above relate to the individual heater sections. When using the sections of the Green heater assembled in hot-blast heating and drying outfits, or for similar purposes, other advantages appear. The individual sections are light, compact and easily handled. Their size and shape permit of their being introduced through narrow doors and stairways. The cost of an air-tight wall on two sides of the heater is saved, since the header covers fit close to one another, making a good joint and leaving only the bottom and top sides of the header stack to be covered or enclosed. If it is desired to protect the headers with a non-conducting covering this can easily be applied in large removable sheets, since, except for the bolts and nuts, the covers present a smooth and even surface.

The sections of Green heaters can be so connected up that some will be using live steam, say while others are using exhaust steam or hot water, the valves being so arranged that individual sections can be thrown from one system to the other at will.

## Efficiency Tests of Steam Pipe Covering

A test of the relative heat losses from bare pipe and from pipe covered with Wyckoff patent steam pipe covering was conducted recently in the Department of Experimental Engineering at Sibley College, Cornell University, by Prof. W. M. Sawdon. The general conditions in both cases were the same. Taking the bare pipe at zero, the results showed a saving in the case of the covered pipe of 83.6%.

The tests consisted in a comparison of the heat radiated by 100 ft. of 1½ in. bare pipe and 100 ft. of 1½ in. covered pipe placed side by side in a room in the west basement of Sibley College. There was but one outside door to the room and it was kept closed to avoid air currents.

As the steam supplied was likely to be wet, it was passed through a separator, then through a riser which was also provided with a drip. The steam then passed through two short lengths of horizontal pipe extending in opposite directions and into the pipes to be tested. The desired steam pressure was maintained partially by the valve on the steam line from the boiler and partially by varying the opening of the separator drip. The pipes and fittings leading from the separator were thoroughly covered with hair felt up to the beginning of the pipes to be tested.

A gauge was used for convenience in regulating the steam pressure for the two high pressure runs. For the low pressure runs a long mercury manometer was used, it being more sensitive and accurate.

The steam condensed in the pipes was caught in a collector, having a glass gauge; it then passed into tanks partly filled with cold water. The tanks were carefully weighed before and after the tests, the difference being the steam condensed.

At the beginning of each test the condensed steam was adjusted to a definite height in the collector glass and maintained at that point, as nearly as possible, throughout the entire run.

Almost three hours was allowed for the pipes and covering to become thoroughly heated and the conditions to become steady before the test was begun.

Observations were taken as shown on the data sheets following.

The temperature of the exterior of pipe covering was measured by nailing a small block, with a hole for a thermometer, to the side of the covering. The thermometer was put in place and the hole filled with dry plaster of paris. The temperature of the outside of the bare pipe was measured by fitting a block of wood with an asbestos packing against the pipe and clamping it firmly. This small block had a hole for the thermometer so that the thermometer bulb would be against the side of the pipe and be surrounded by mercury.

The temperature of the steam was obtained by placing thermometers in cups extending into the steam where it entered and also where it left the pipes being tested. These cups were filled with cylinder oil.

The air temperature in the vicinity of the bare pipe and for the covered pipe was also taken by thermometers. The quality of the steam was found by a throttling calorimeter for each side, and it was found to agree almost exactly.

The condensation taking place in the collectors was found by separate runs with pressures corresponding to those maintained during the covering tests. For this purpose the covered pipe was removed and the collector put in its place. This condensation was found to be equivalent to 1.6 B. T. U. per degree difference of temperature per hour, and this correction was applied to the results.

The scales used for weighing the condensed steam were graduated to 1/100 lb., and were quite sensitive, while the thermometers were calibrated in a bath of oil by referring to a standard, and this correction applied.

The efficiency was found by dividing the difference between the B. T. U. radiated by the bare pipe and the covered pipe by the B. T. U. radiated by the bare pipe.

Following is a summary of two of the tests:

TEST No. I

Duration of Run, 2½ hrs. Barometer, 29.37″. Size of Pipe, 1½″. Length of Pipe, 100 ft. Radiating surface, 56.2 ft.

| | Bare Pipe | Covered Pipe |
|---|---|---|
| Steam pressure, absolute, lbs. | 24.5 | 24.5 |
| Quality of steam | 99.8 | 99.8 |
| Temperature of steam, °F | 239 | 239 |
| Temperature of air, °F | 82 | 80 |
| Temperature diff., air and steam, °F | 157 | 159 |
| Temperature exterior of pipe and covering, °F | 234 | 91 |
| Wt. of condensed steam, actual | 61.8 | 12.32 |
| Wt. of condensed steam, per hr., dry, lbs. | 24.7 | 4.93 |
| B. T. U. radiated per lb. steam | 946.8 | 946.8 |
| B. T. U. radiated per hr. | 23386 | 4668 |
| B. T. U. radiated by collector per hour | 251 | 254 |
| B. T. U. radiated by pipe, net | 23135 | 4414 |
| B. T. U. radiated per deg. diff. temp., air and steam, per hr | 147.4 | 27.76 |
| B. T. U. radiated per deg. diff. temp., air and steam, per hr., per sq ft. | 2.62 | .494 |
| Relative amount of heat transmitted, % | 100 | 18.8 |
| Efficiency of covering | | 81.2 |

TEST No. II

Duration of Run, 3 hrs. Barometer, 29.4″. Size of Pipe, 1½″. Length of Pipe, 100 ft. Radiating Surface, 56.2 sq. ft.

| | Bare Pipe | Covered Pipe |
|---|---|---|
| Steam pressure, absolute, lbs. | 107.7 | 107.7 |
| Quality of steam | 99.5 | 99.5 |
| Temperature of steam, °F | 333 | 333 |
| Temperature of air, °F | 87 | 83 |
| Temperature difference, air and steam, °F | 246 | 250 |
| Temperature exterior of pipe covering | 325 | 104 |
| Wt. of condensed steam, actual, lbs. | 151.27 | 26.14 |
| Wt. of condensed steam, per hour, dry | 150.4 | 26 |
| B. T. U. radiated per lb. steam | 880 | 880 |
| B. T. U. radiated per hour | 44117 | 7627 |
| B. T. U. radiated by collector, per hour | 394 | 400 |
| B. T. U. radiated by pipe per hour, net | 43723 | 7227 |
| B. T. U. radiated per deg. diff., temp., air and steam, per hr | 177.7 | 28.9 |
| B. T. U. radiated per deg. diff., temp., air and steam, per hr., per sq. ft. | 31.6 | .514 |
| Relative amount of heat transmitted, % | 100 | 16.3 |
| Efficiency of covering, % | | 83.7 |

## The Weather for December

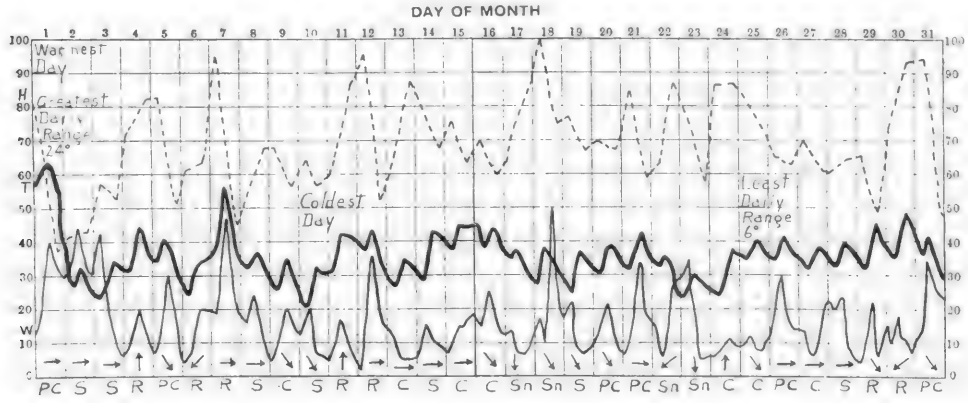
The mild weather conditions noted in October and November continued during December, bringing the temperature figures for the first part of the winter of 1908-1909 considerably above those of most other years. All of the five cities from which detailed reports are received experienced mean temperatures above normal.

NEW YORK

Highest temperature during the month, 64° on the 1st; lowest temperature, 20° on the 10th; greatest daily range, 24° on the 1st; least daily range, 6° on the 25th. Mean temperature for the month, 35°, 1° above the average for 38 years. Total precipitation for the month, 3.21 in., a trifle below normal. Total snowfall, 5.1 in. No snow on ground at end of month. Prevailing direction of the wind, from the west; total wind movement, 10,712 miles; average hourly velocity, 14.4 miles. There were 7 clear days, 9 partly cloudy and 15 cloudy. Rain or snow fell on 10 days.

CHICAGO

Highest temperature, 53° on the 29th; lowest temperature, 10° on the 8th; greatest daily range, 26° on the 30th; least daily range, 5° on the 17th. Mean temperature for the month, 31°, 2° above the average mean of this month for 38 years. Total precipitation for the month, 1.18 in., nearly 1 in. below normal. Total snowfall, 2.7 in. No snow on ground at end of month. The prevailing direction of the wind was west; total movement, 12,064 miles; average hourly velocity, 16


DAY OF MONTH
Warmest Day
Greatest Daily Range 24°
Coldest Day
Least Daily Range 6°
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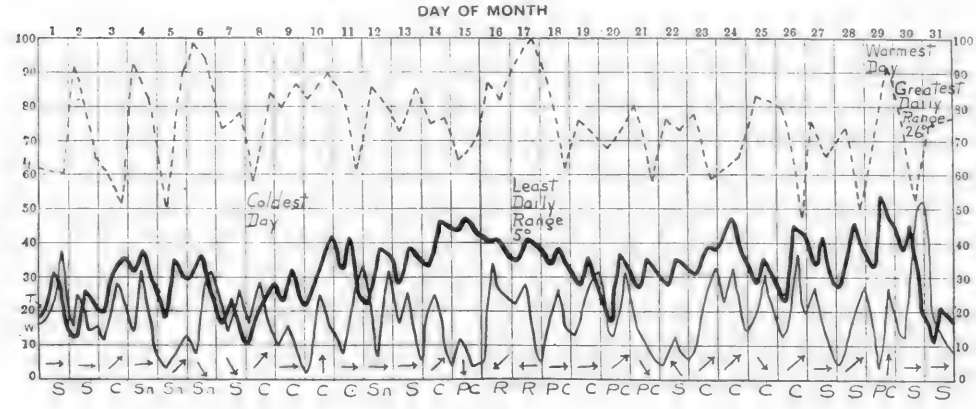
RECORD OF THE WEATHER IN NEW YORK FOR DECEMBER, 1908


DAY OF MONTH
Warmest Day
Greatest Daily Range 26°
Coldest Day
Least Daily Range 5°
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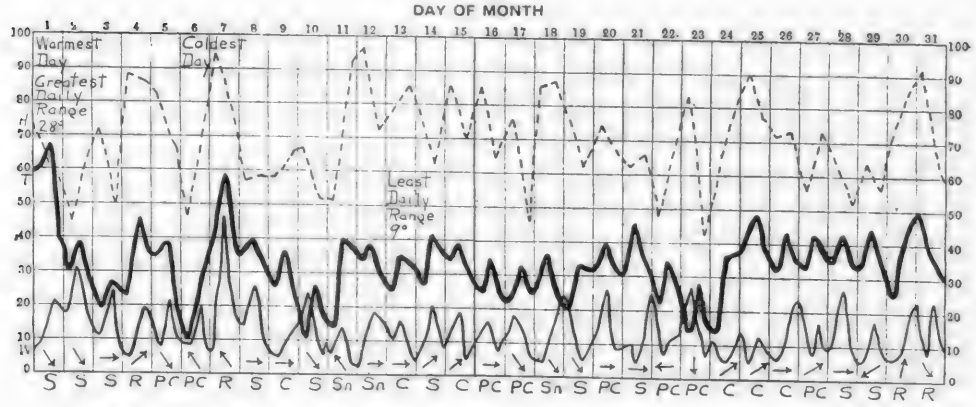
RECORD OF THE WEATHER IN CHICAGO FOR DECEMBER, 1908

Plotted from records especially compiled for THE HEATING AND VENTILATING MAGAZINE, by the United States Weather Bureau

Upper line indicates temperature in degrees Fahr.
Broken lines indicates relative humidity in percentage from readings taken at 8 a. m. and 8 p. m.
Lower line indicates velocity of wind in miles per hour.
Arrows fly with prevailing direction of wind.
S—Clear. PC—Partly Cloudy. C—Cloudy. R—Rain. Sn—Snow.


DAY OF MONTH
Warmest Day
Coldest Day
Greatest Daily Range 28°
Least Daily Range 9°
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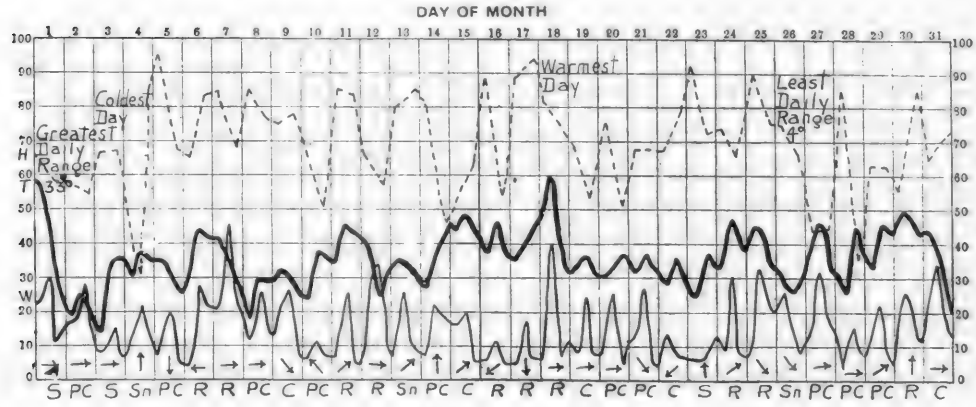
RECORD OF THE WEATHER IN BOSTON FOR DECEMBER, 1908


DAY OF MONTH
Coldest Day
Warmest Day
Least Daily Range 4°
Greatest Daily Range 35°
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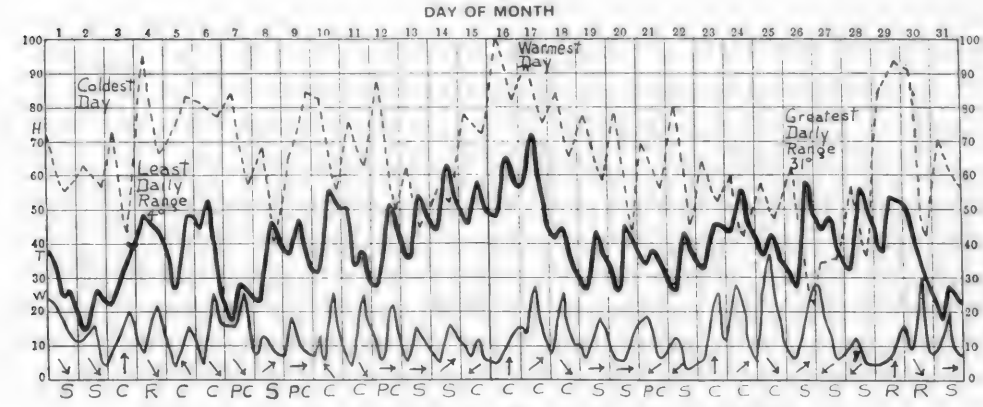
RECORD OF THE WEATHER IN PITTSBURG FOR DECEMBER, 1908


DAY OF MONTH
Coldest Day
Warmest Day
Least Daily Range 4°
Greatest Daily Range 31°
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S S C R C C PC S PC C C PC S S C C C C S S PC S C C C S S S R R S

RECORD OF THE WEATHER IN ST. LOUIS FOR DECEMBER, 1908

Plotted from records especially compiled for THE HEATING AND VENTILATING MAGAZINE by the United States Weather Bureau

Upper line indicates temperature in degrees Fahr.
Broken line indicates relative humidity in percentage from readings taken at 8 a.m. and 8 p.m.
Lower line indicates velocity of wind in miles per hour.
Arrows fly with prevailing direction of wind.
S—Clear. PC—Partly Cloudy. C—Cloudy. R—Rain. Sn—Snow.

miles. There were 9 clear days, 8 partly cloudy and 16 cloudy. Rain or snow fell on 7 days.

BOSTON

Highest temperature, 67° on the 1st; lowest temperature, 10° on the 6th; greatest daily range, 28° on the 1st; least daily range, 9° on the 13th. Mean temperature for the month, 33°, 1° above normal for 38 years. Total precipitation for the month, 2.47 in., about 1 in. below normal. Total snowfall, 3.5 in. No snow on ground at end of month. Prevailing direction of the wind, from the west; total movement, 8,457 miles; average hourly velocity, 11.4 miles. There were 10 clear days, 9 partly cloudy and 12 cloudy. Rain or snow fell on 7 days.

PITTSBURG

Highest temperature, 60° on the 18th; lowest temperature, 14° on the 3rd; greatest daily range, 33° on the 1st; least daily range, 4° on the 26th. Mean temperature for the month, 35°, 1° above the mean of this month for 38 years. Total rainfall, 1.94 in., nearly 1 in. below normal. Total snowfall, 1.3 in., of which a trace remained at end of month. Prevailing direction of the wind, west; total movement, 9,138 miles; average hourly velocity, 12.3 miles. There were 3 clear days, 11 partly cloudy and 17 cloudy. Rain or snow fell on 13 days.

ST. LOUIS

Highest temperature, 72° on the 17th; lowest temperature, 14° on the 2nd; greatest daily range, 31° on the 26th; least daily range, 7° on the 4th. Mean temperature for the month, 39°, 3° above the mean of this month for 36 years. Total rainfall, 0.64 in. There was only a trace of snow. The average precipitation of this month for 38 years is 2.19 in. Prevailing direction of the wind, northwest; total movement, 7,998 miles; average hourly velocity, 10.8 miles. There were 13 clear days, 4 partly cloudy and 14 cloudy. Rain fell on 4 days.

## Current Heating and Ventilating Literature

*Under this heading is published each month an index of the important articles on the subject of heating and ventilation that have appeared in the columns of our contemporaries. Copies of any of the journals containing the articles mentioned may be obtained from* THE HEATING AND VENTILATING MAGAZINE *on receipt of the stated price.*

Modern methods of heating. H. R. Basford. Read before the Pacific Coast Gas Assn. Gives a report of automatic gas water heating devices and their performances. Discussion. 4,400 w. pro. Age—Nov. 16, 1908. 20c.

POWER PLANTS

Power plant of the New Plaza Hotel, New York. Plans and detailed description of a mechanical equipment of unusual size. 5,000 w. Eng. Rec.—Nov. 21, 1908. 20c.

TURBINES

The development of the Small Steam Turbine. Charles A. Howard. Illustrates and describes features of design and construction of leading types of small capacity turbines used in the United States. 2,500 w. Eng. Mag.—Dec., 1908. 40c.

HOT WATER HEATING

The Design of Low-Pressure Hot-Water Heating Systems (Calcul des Installations de Chauffage par l'Eau Chaude à Basse Pression). E. Mathieu. A mathematical discussion. Ills. Serial. First part. 1,000 w. L'Hygiene du Batiment-June, 1908. 40c.

Second part. 1,000 w. July, 1908. 40c.

Third part. 1,000 w. August, 1908. 40c.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

## Program for Annual Meeting

Following is the program of the professional as well as the social sessions of the annual meeting of the society, held in the Engineering Societies Building, 29 West 39th Street, New York, Tuesday, Wednesday and Thursday, January 19, 20 and 21, 1909:

**Tuesday Afternoon, January 19, 2 P. M.**

Reports of officers and regular committees.

**Tuesday Evening, January 19, 8 P. M.**

Report of tellers, announcing names of newly elected officers.

Report of committee to collect data on fan blast heating, William G. Snow, chairman.

Report of committee to collect data of hot water heating for residences, E. F. Capron, chairman.

Report of committee to collect data on the relative corrosion of wrought iron and steel pipes, Thomas N. Thomson, chairman.

Report of committee on the rating of house heating boilers, J. H. Kinealy, chairman.

Paper, "The Rating of House Heating Boilers," by Prof. William Kent.

TOPICS FOR DISCUSSION

1. A standard rating for fans and blowers, based on diameter and width of wheel and revolutions per minute.

2. The necessity of an accurate standard rating for house heating boilers.

**Wednesday Afternoon, January 20, 2 P.M.**

Paper, "Air Cooling by Refrigeration," by W. W. Macon.

Paper, "Notes on the Design of Furnace Heating Systems, with an Application to a 10-Room Residence," by Prof. J. D. Hoffman.

TOPICS FOR DISCUSSION

3. Air cooling by means of air washers.
4. The relative carrying capacity of horizontal and vertical warm air pipes.
5. The effect of shape on warm air risers.
6. The duty of the society relative to informing the pulic of the lack of ventilation in halls and places of pulic gatherings, particularly in the thousands of storerooms fitted up for moving picture shows, where young people are subject to enervation and contamination.
7. The necessity for having ventilation laws applied to the installation of apparatus in old as well as new buildings.
8. State control for heating and ventilating engineers.

TOASTS AND SPEAKERS AT SOCIETY'S ANNUAL DINNER, NEW GRAND HOTEL, BROADWAY AND 31ST STREET, WEDNESDAY EVENING, JANUARY 20, 7:30 P. M.

Toastmaster, President James Mackay.
The Engineer Abroad. Letters from our foreign members.
"The Position of the Heating and Ventilating Engineer Before the Public," D. D. Kimball.
"Opportunities," Reginald Pelham Bolton.
"The Society's Aim," Henry L. Doherty.
"The Society's Influence," S. A. Jellett.
"Future of the Society," William G. Snow.

### Thursday Morning, January 21, 10:30 A. M.

Paper, "Miniature Steam Traps and Their Application to Steam Circulation," by James A. Donnelly.

TOPICS FOR DISCUSSION

9. The value of tabulated experience recognized as science compared with individual opinion and experience.
10. A table or sliding scale of indoor and outdoor temperatures for gauging the capacity of heating plants to maintain a 70° temperature inside, with the outside temperature at zero, this to be established by the society, so that it could be widely incorporated in heating specifications and add to the society's prestige.
11. The maximum permissible length for a steam main in a gravity apparatus operating under a pressure not to exceed 2 lbs. on one and two pipe systems.
12. The relative area of wet and dry return mains in a gravity steam heating apparatus.
13. The deposit of lime or other mineral from impregnated water in a heated pipe, with relation to diameter or area, or the relative deposit in a 1 and 2 in. pipe in a given time.

### Thursday Afternoon, January 21, 2 P. M.

Paper, "The Effect of Wind on Heating and Ventilating, by H. W. Whitten.
Installation of new officers.

TOPICS FOR DISCUSSION

14. A standard schedule of blast area and linear feet of 1 in. pipe in air heating stacks and coils, based on the cubic feet of air passed through the heater per hour or minute, and the mean difference between inside and outside temperature at velocities from 1,000 to 1,800 ft. per minute.
15. The factor to be taken into consideration and the proper method of rating fan heating coils in determining the coefficient of heat transmission under different conditions of temperature.
16. The standard rules of the society for efficiency and value of the heating surface of different types of direct, direct-indirect, indirect and forced blast radiation.

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### Arrangements for Annual Dinner

Tickets for the annual dinner of the society, to be held in the New Grand Hotel, Broadway and 31st Street, Wednesday evening, January 20, at 7:30 o'clock, may be obtained from the secretary, W. M. Mackay, at $2.50 each. Immediately preceding the dinner, from 6 to 7:30 P. M., a general reception of members, ladies and guests will be held in the parlors of the New Grand Hotel. A special porlor will be provided at the hotel for the use of the ladies during the entire meeting.

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### Important Conference on the Ratings of House Heating Boilers

Following several preliminary conferences and overshadowing in importance any other one feature of the program of the American Society of Heating and Ventilating Engineers, a meeting has been arranged for Tuesday evening, January 19, in the rooms of the Engineering Societies Building, between the heating engineers' society, and committees representing the Boiler and Radiator Manufacturers' Association and the National Association of Master Steam and Hot Water Fitters. The subject under discussion, a proposed revision of the ratings of house heating boilers, is recognized as one of the most important and troublesome questions now before the heating profession for solution. The matter has been agitated for years, but at no time have the prospects been so bright for a satisfactory outcome as they are at present.

The committee representing the master steam fitters is made up of E. M. Nichols,

Philadelphia, chairman; Burton P. Hall, New York and T. B. Cryer, Newark. In addition to this committee, it is expected that the board of directors of the master fitters' association, which was called to meet in New York January 19, will accept the heating engineers' invitation to be present at the conference.

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The following candidates for membership in The American Society of Heating and Ventilating Engineers have been elected:

NEW MEMBERS

Louis K. Berman, manager, Raisler Heating Co., 1966 Broadway, New York.

M. M. Cochran, manager heating department, Crane & Ordway Co., St. Paul, Minn.

Charles A. Gillham, secretary and general manager, Central Station Engineering Co., State Bank Bldg., Lebanon, Ind.

Joseph G. Hayes, vice-president and manager, Hayes Brothers, Inc., Indianapolis.

John H. Kassa, sales manager, Bayley Mfg. Co., Milwaukee.

George W. Knight, consulting engineer, Board of Education, Newark, N. J.

Arthur H. Leitch, traveling representative engineer, Sheldon's Ltd., Galt, Ont.

W. W. Macon, managing editor, "The Metal Worker," New York.

Peter F. Maginn, Peter F. Maginn & Co., 1025 Forbes Street, Pittsburg, Pa.

Sidney R. Sheldon, chief engineer, Sheldon's Ltd., Galt, Ont.

Lawrence C. Soule, salesman and engineer, Vento Blast Heater Department, American Radiator Co., Chicago.

Frank H. Stevens, engineer, American District Steam Co., Monadnock Block, Chicago.

John H. Textorius, engineer, Johnson Heating Co., 138 East Thirty-first Street, New York.

Jacob O. Trautwein, engineer, L. E. Rodgers Engineering Co., Chicago.

James R. Wade, president, National Vacuum Steam Heating Co., St. Louis.

ASSOCIATE MEMBER

Hubert R. Hill, manager heating department, W. W. Montague & Co., San Francisco.

NATIONAL ASSOCIATION OF MASTER STEAM AND HOT WATER FITTERS

### Dinner of Newark (N. J.) Local Association

A large and representative gathering of the heating trade in Newark and vicinity sat down to the fourth annual banquet of the Master Steam and Hot Water Fitters' Association of Newark, held December 9, in Davis's Restaurant, Newark. The dinner was enlivened by the strains of a capable orchestra and several soloists, while the speakers testified to the healthy condition of the Association in that important section. Among the representatives of the supply trade were noticed the following:

E. O. Haskins, Abendroth Bros.; W. F. Kraft, Richardson & Boynton Company; W. A. Birdsall, Model Heating Company; J. J. Blackmore, J. L. Mott Iron Works; R. D. Elder, Ronalds & Johnson Company; A. L. DeVoe, Thatcher Furnace Company; James E. Dobbs and J. Barton Garfield, Pressed Radiator Company; C. F. Young, A. H. Magee Company; W. A. Russell, W. A. Russell & Co.; D. W. Jones, Crane Company; C. E. Gorton, Gorton & Lidgerwood Company; F. E. Sass, Jenkins Bros.; W. P. Davenport, Wm. H. Page Boiler Company; F. W. Jaqui, Boynton Furnace Company; G. C. Hall, H. W. Johns-Manville Company, and Chas. S. Wood, Keasbey & Mattison Company.

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### American Institute of Chemical Engineers

This association, organized during the past year, held its first annual meeting in Pittsburg, December 28-29, 1908. Sessions were held at the Carnegie Technical Schools and were notable for the high standard of the papers presented, and the variety of subjects discussed. The program included the following papers having a bearing on the work of the heating and ventilating engineer:

"Steam Power Plant Economics from the Standpoint of the Chemical Engineer," by William M. Booth.

"Technical Coal Analysis," by Edward Gudeman.

"The Chemical Aspect of Impurities which Cause Scale and Corrosion of Steam Boilers," by J. C. William Greth.

"Testing and Performance of Steam Generating Apparatus," by A. Bement.

"The Examination of Flue Gases in Boiler Tests," by H. August Hunicke.

"Dryer Calculations and Dryer Design," by William M. Grosvenor.

The following officers were elected for the ensuing year: President, Samuel P. Sadtler, Philadelphia; vice-presidents, Charles F. McKenna, New York; H. A. Hunicke, St. Louis, Mo.; and Edward G. Acheson, Niagara Falls, N. Y.; secretary, John C. Olsen, Polytechnic Institute, Brooklyn, N. Y.; treasurer, William M. Booth, Syracuse, N. Y.; auditor, Richard K. Meade, Nazareth, Pa.; three directors to serve three years, Ludwig Reuter, Berkeley, Cal.; Thorn Smith, Isabella, Tenn.; and H. F. Brown, Wilmington, Del.

### American Society of Mechanical Engineers

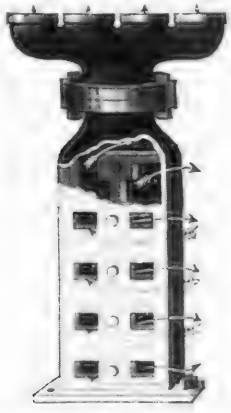
An attendance of 738 members was noted at the recent annual meeting of the American Society of Mechanical Engineers, held in New York December 1-4, 1908. This is the largest attendance of members in the history of the society. The total registration was 1,048, which is not as high as that of a year ago, but it actually represents a larger attendance as this year only those were registered as guests who were to participate in the social functions. The following officers were elected for the ensuing year:

President, Jesse M. Smith, New York; vice-presidents, George M. Bond, Hartford, Conn.; Prof. R. C. Carpenter, Ithaca, N. Y.; and F. M. Whyte, New York; Managers, H. L. Gantt, Pawtucket, R. I., Will J. Sando, Milwaukee, Wis.; and I. E. Moultrop, Boston; treasurer, Major William H. Wiley, New York.

The society reports a total membership 3,455, made up of 15 honorary members, 2,322 regular members, 357 associate members and 761 junior members.

### Detachable Water Post of Spence Hot Water Boiler

The accompanying illustration shows the detachable water post of the Spence hot water boiler, already described in these columns. This device, which is an exclusive one, found only on the Spence boiler, is so constructed that the only water connection is between the water post and sections on the outside of the



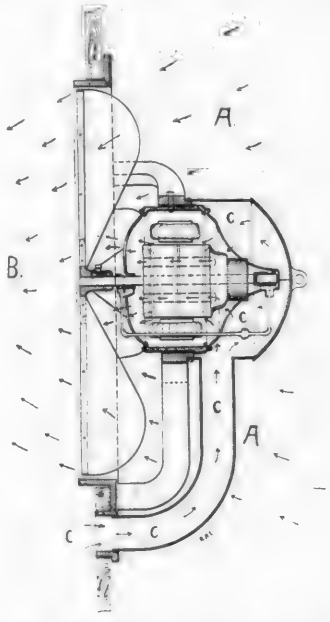
DETACHABLE WATER POST OF THE SPENCE BOILER

boiler. Consequently erection is easy and there can be no leaky joints as the water connections are made with genuine "Rainbow" packing. With the detachable water post, each section is an independent heater. It allows for using one section for heating water for domestic purposes or, in case of accident to any one of the sections, it can be blanked at the water post and the boiler continued in operation.

A Blue Book describing the boiler in detail, may be had by addressing the manufacturers. Pierce, Butler & Pierce Mfg. Co., Syracuse, N. Y.

### A Fully Enclosed Ventilated Type Motor Fan

An electric direct connected exhaust fan, in which the motor is kept cool by the fan itself, is an interesting device manufactured by the Ilg Electric Ventilating Co. of Chicago and 145 Chambers




ELECTRIC EXHAUST FAN WITH FULLY ENCLOSED MOTOR, THE MOTOR BEING COOLED BY THE FAN ITSELF

Street, New York, and shown in the accompanying illustration.

The motor is fully enclosed as shown and is thus protected against smoke or foreign matter. The cooling of the motor is accomplished by means of a small tube through which the air is drawn from the outside by the fan. This effectually keeps the motor cool at all times and under varying conditions, eliminating the objectionable feature of the heating of the armature, a condition often met with in connection with enclosed type motors used in connection with ventilating fans. This fan is more fully described in Bulletin H. V. No. 20, issued by the company, and which, we understand, may be obtained either from the New York or Chicago office.

### The Kinealy Air Purifier and Humidifier

The accompanying illustrations are among the first that have been published of the Kinealy air purifier and humidifier, invented by Prof. J. H. Kinealy, of St. Louis. The apparatus may be so located that the air is either drawn or forced through it by the fan. The spray heads

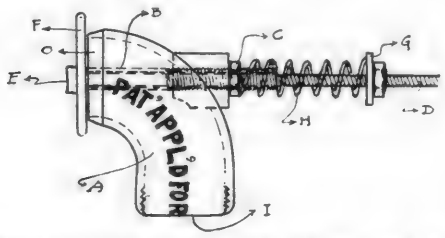



FIG. 1.—DESIGN OF SPRAY HEAD OF KINEALY AIR PURIFIER

are of unique construction, and are especially designed to be easily flushed or cleaned. By adjusting the tube B, shown in Fig. 1, so as to secure the desired opening through which the water must pass between the spray plate F and the end of the outlet O, a spray of any fineness may be obtained. By the construction of the spray head, it is possible to secure water both in a sheet form and in a finely-divided mist or spray. When it is desired to flush the spray head, the rod D is pushed forward so as to make the head E move away from the outlet O. The pressure of the water acting on the spray plate F moves it away from the outlet so that the opening through which the water can pass is enlarged. When the pressure on the end of the rod is removed, the spring H expands and presses on the washer G and moves the rod back to its original position.

A device that flushes the spray heads automatically every 30 seconds is included in the apparatus, shown in Fig. 2. This device is operated by means of a swinging frame B acting on the rod D of each of the spray heads. The movement is secured through the tilting of the water pan C, which falls forward, as indicated by the dotted lines, when it accumulates a certain amount of water, causing the hammer H to press the frame against the rods D. The apparatus is manufactured by the Kauffman Heating & Engineering Co., 2320 Olive street, St. Louis, Mo.

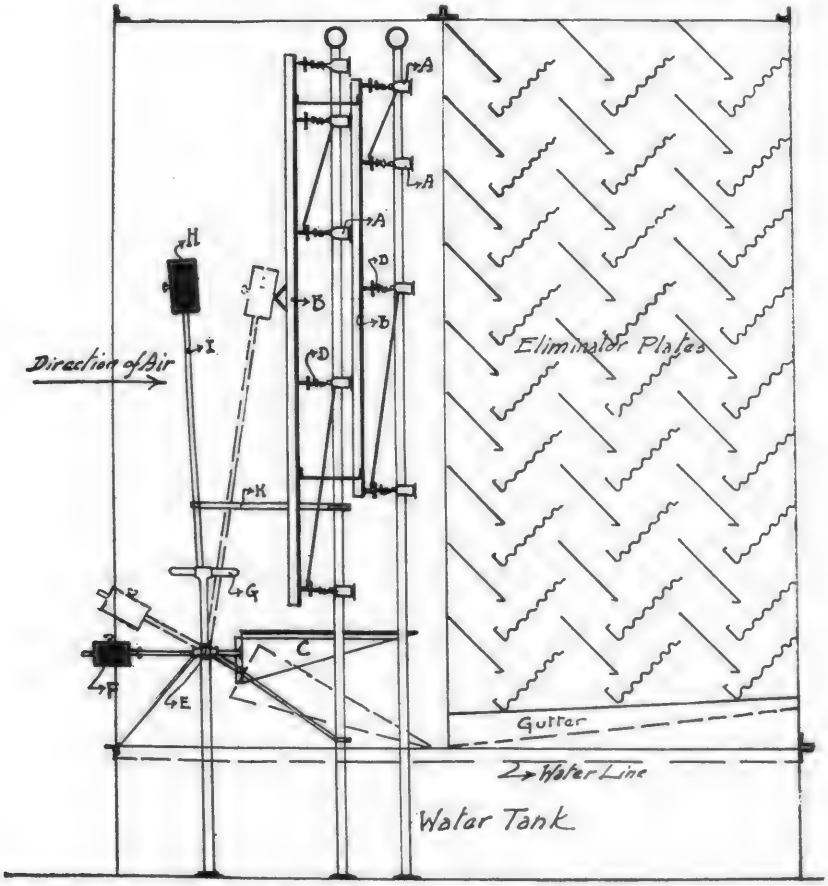



FIG. 2.—SECTIONAL ELEVATION OF KINEALY AIR PURIFIER, SHOWING AUTOMATIC FLUSHING DEVICE FOR SPRAY HEADS

# TRADE AND MISCELLANEOUS NOTES

## Deaths

**Edward F. C. Young,** president of the Joseph Dixon Crucible Co., Jersey City, N. J., died December 6, at his home in Jersey City. He was in his 74th year. Mr. Young was connected with a number of the largest financial institutions in New Jersey. He leaves a widow, one son and one daughter.

**Warren N. Reigel,** secetary of the Penn Engineering Co., Philadelphia, died in that city December 9, of heart failure. Mr. Reigel was for 10 lears secretary of the company and had been connected with the firm since its incorporation in 1894.

**Fletcher W. Rockwell,** first vice-president of the National Lead Co., and manager of the Chicago branch, died in that city December 12. He was 65 years old.

## Miscellaneous Notes

**Ira Budd,** Newark, N. J was tendered a dinner December 16 by his associates in the heating and plumbing trade in honor of his election as a member of the Board of Works of Newark. Among speakers at the dinner, were A. Selden Walker, Louis Maier, Alfred N. Dalrymple, chairman of the Repulican County Committee, and Agustave Eggers, commissioner of the Board of Works.

**Waukesha, Wis.**—The Waukesha Gas & Electric Co. has been granted an extension of time by the city council until December 1, 1909, in which to construct its proposed central heating plant.

**Little Rock, Ark.**—The receivership of the Central Heating & Mfg. Co has been terminated. H. M. Armistead, the receiver appointed five months ago, will continue to have charge of the corporation's affairs. The new directors are Judge J. M. Spradling of Hartford and Little Rock, who is also the new president of the company; E. T. Reaves, George G. Worthen, T. V. Wilson and Judge W. M. Kavanagh, president of the Southern Trust Co. It is proposed to enlarge the plant and add a number of new features.

**Washington, D. C.**—As the result of the recent deaths of three persons, caused, it is supposed, by the inhaling of carbon

ENGINEERS AND STEAMFITTERS REQUIRE GOOD TOOLS

The GENUINE ARMSTRONG Stocks and Dies are

RELIABLE, ACCURATE, EASY WORKING

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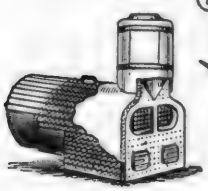

ROBERT A. KEASBEY CO.

HEAT AND COLD INSULATING MATERIALS

HEADQUARTERS FOR
85% MAGNESIA, ASBESTOS AND BRINE
PIPE COVERINGS, ASBESTOS PRODUCTS, ETC.

ESTIMATES FURNISHED AND CONTRACTS EXECUTED.

100 NORTH MOORE ST, NEW YORK CITY.

monoxide produced by a gas water heater. Henry B. Davis, chief inspector of plumbing, has recommended an amendment of the gas fitting regulations, providing that every gas water heater installed in Washington will be without a pilot light, which will make it necessary to open the door of the heater to light the gas. The heater also must be provided with a flue to the outer air according to the proposed regulation. It has been found that when a constant stream of cold water is passing through the coils the evolution of carbon monoxide is greatly increased.

**A. Burlingame Co.,** Worcester, Mass., jobber of heating and plumbing supplies, suffered a loss by fire December 8, amounting to $40,000.

**Professor William Kent,** formerly dean of the L. C. Smith School of Applied Science, Syracuse University, Syracuse, N. Y., has been appointed general manager of the Sandusky Foundry & Machine Co., Sandusky, Ohio.

**Madison, Wis.**—A law was passed at the last session of the state legislature authorizing each rural school district to spend $50 for the ventilation of school houses. As there are 4,000 district schools in the state, the amount of money the state will pay out in accordance with the provisions of this law, is $200,000. There is much speculation as to the purpose of the law, as country schoolhouses as a rule are notoriously well ventilated through ill fitting windows, broken floors and worn out doors.

**George D. Howell,** vice-president of the McCrum-Howell Co., New York, was the host at a dinner at the Duquesne Club, in Pittsburg, December 3, on the occasion of the meeting in that city of the American Mining Congress. Among the guests were Secretary of the Interior J. B. Garfield, Government Expert in Charge Holmes, Judge Richards, president of the Mining Congress; Judge Buffington, Congressmen A. F. Cooper and Englesbraith of California, Senator Dick of Ohio, Messrs. Zerbe and F. M. Osborne of Cleveland, Julian Kennedy of the Orient Coke Co.; John Mitchell, the labor leader; R. C. Crawford, of the Emerald Coal Co.; W. Clyde Wilkins, trustee of the Thaw Estate; and S. A. Taylor.

### Manufacturers' Notes

**Utica Heater Co.,** Utica, N. Y., has increased its capital stock from $100,000 to $175,000, the increase being classified as first preferred stock.

**Crane Co.,** Chicago, distributed among its employees at Christmas a bonus of

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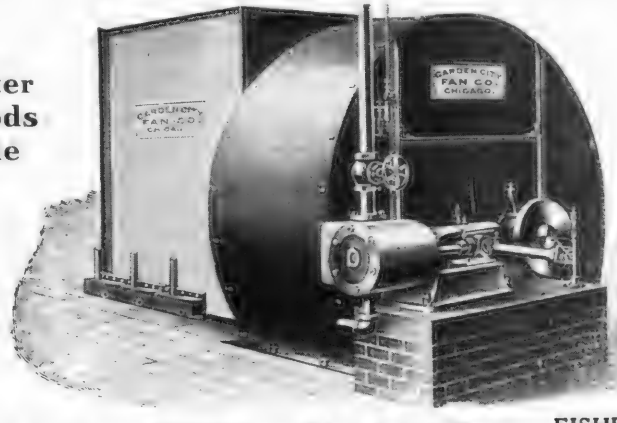


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10% of their yearly wages. The sum thus apportioned amounted to $350,000.

**McCrum-Howell Co.**, New York, will increase the facilities of its plant at Norwich, Conn., by the addition of the warehouse 50 x 200 ft.

**John T. Young Boiler Co.**, Norwich, Conn., has secured land in that place on which it will build a boiler plant.

**Kellogg-Mackay-Cameron Co.**, Chicago, announces the removal of its Chicago offices and showrooms to its warehouse at Eighteenth Street and Stewart Ave.

**Ohio Seamless Tube Co.**, Shelby, O., will start up its new plant about February 1. The buildings, entirely of steel framework construction, have been built on the site of the Shelby Seamless Tube Co., which was destroyed by fire in June last.

**Allen, Myers & Co.**, Rock Island, Ill., jobbers of heating and plumbing supplies, are rebuilding on the site of their quarters recently destroyed by fire.

**Lunkenheimer Co.**, Cincinnati, O., was a successful bidder for bronze globe and angle valves for the Isthmian Canal Commission for use in connection with the Panama Canal. The firm has received an order for 7,000 valves, ranging from ¼ in. to 3 in., the type used being the Lunkenheimer Renewo renewable seat and disc regrinding valve, guaranteed for 200 lbs. working presure.

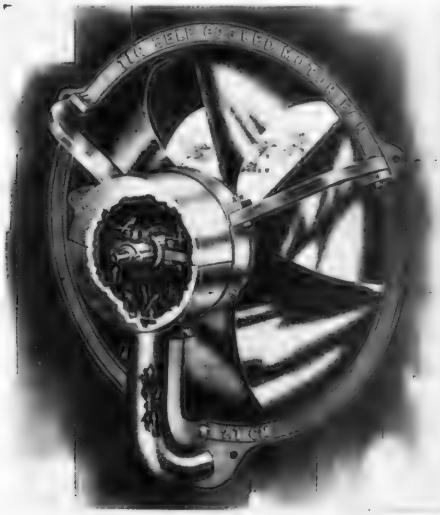
**Kellogg-Mackay-Cameron Co.**, Chicago, will build a four-story reinforced concrete warehouse at Ninth Avenue and Fourth Street, South, Minneapolis, on a sidetrack of the Chicago, Milwaukee & St. Paul Railway. In addition to the main building, there will be built a large pipe shed.

**A. J. Lindemann & Hoverson Co.**, Milwaukee, Wis., has moved into its new building at First and Russells Avenue. A salesroom is located at 497 East Water Street, in charge of John S. Brennan, treasurer of the company.

**Utica Heater Co.**, Utica, N. Y., has opened a branch office and warehouse at 105 Lake Street, Chicago, in charge of Fred B. Edell.

**American Blower Co.**, Detroit, Mich., distributed a bonus to its employees at Christmas, each employee receiving $1.00 and $1.00 additional for each year of continued employment by the company.

**Fowler & Wolfe Mfg. Co.**, Philadelphia, in a New Year's card of greeting to the trade announces that immediately after January 15, 1909, when accurate costs can be determined, new quotation sheets will be issued, the prices being as low as is consistent with the high

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grade material, workmanship, etc., that is usual with the Fowler & Wolfe wall radiation.

### Business Changes and New Firms

**Sawnie-Miller Co.**, Nashville, Tenn., has opened an office at 156 Fourth Avenue, North, as contractors in steam and hot water heating. The principals of the firm are H. M. Sawnie, formerly vice-president of the Jones & Hopkins Mfg. Co. and W. J. Miller, of Miller & Co., grain dealers, both of Nashville.

### Contracts Awarded

**H. Sweeney & Co.**, Seattle, Wash., heating plant for old county hospital, to cost $3,850.

**Idaho Plumbing & Hardware Co.**, Boise, Idaho, heating, ventilating and plumbing new east wing of Boise high school building to be built early in the spring. The contract amounts to $11,786.

**John W. Danforth Co.**, Buffals, N. Y., mechanical equipment for new round house and shops of the N. Y. C. & H. R. Railroad at Rensselaer, N. Y. Four 300 H. P. water-tube boilers will be installed. The contract also includes a vacuum system to cost $4,000.

**Henshaw, Bulkley & Co.**, San Francisco, Cal., heating and ventilating new Phelan building at O'Farrell and Market Streets, San Francisco, for $11,300.

**M. B. Sanders**, Little Rock, Ark., heating and plumbing new addition to the government building for $7,000.

**American Warming & Ventilating Co.**, Pittsburg, Pa., heating two new school buildings at Buckhannon, W. Va., for $7,000.


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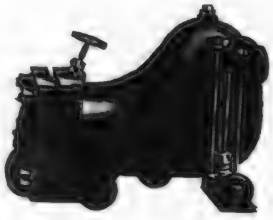

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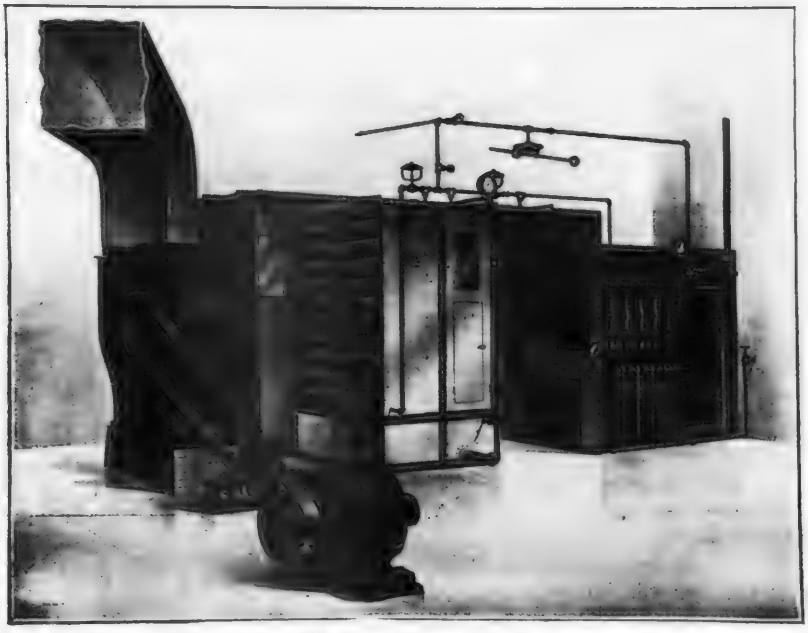

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**Pittsburgh Gage & Supply Co.,** Pittsburg, Pa., high pressure steam piping for Provident Coal Co., St. Clairsville, O.; for the W. B. Skelley Coal Co., at Export, Pa.; for the new Allegheny General Hospital on the North Side, Pittsburg, and for the Monongahela Incline Co., Pittsburg.

**Norman S. Kellogg,** central heating plant for school buildings in Montclair, N. J. Contract price, $23,763. The matter of the contract was taken before the Supreme Court in Trenton and the job has been held up on the ground that the plans for the heating plant had not been approved by the state board and "that the specifications were so drawn that it practically shut out competitive bidding."

**Tholen Bros.,** Leavenworth, Kan., steam heating government buildings at Fort Riley. The contract amounts to $20,000.

**Iron City Heating Co.,** Pittsburg, Pa., heating plant for new $300,000 Y. M. C. A. in the East End.

**H. W. Moffit & Co.,** San Francisco, Cal., heating and ventilating new Mission grammar school for $13,300.

---

**Pipe and Boiler Insulation** is the title of a new 72-page catalogue which has just been issued by the H. W. Johns-Manville Co., of New York. This book is devoted to a presentation of the problems of insulating all kinds of heated and cooled surfaces, such as pipes, boilers, furnaces, flues, ducts, etc., as well as insulation for refrigerating and cold storage work. The book is the most complete one of its kind we have seen and bears out the company's idea, "coverings for every condition." It is very handsomely illustrated and printed throughout. We understand that a copy of the catalogue can be secured by addressing the company's nearest branch and asking for Catalogue, No. 100.

**Inside Facts About Hot Blast Heater Coils,** a 16-page pamphlet published by the Green Fuel Economizer Co., Matteawan, N. Y., describes a new type of hot-blast heater coil or radiator for indirect heating systems.

The novel feature of this heater is that the circulation of steam, air and condensation is positive and continuous, and in the natural direction of gravity drainage from inlet to outlet, whereas, in heaters employing a single header at the bottom the air and water must at times circulate either against the current of steam or else contrary to the force of gravity. Other advantages of this new heater are accessibility to each individual pipe for inspection, cleaning or repair without disturbing other pipes and the ability to circulate hot or cold water or brine, if desired. The several novel features of construction described and illustrated in this booklet will be of interest

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to engineers who are called upon to design, purchase, or install hot-blast heating and drying outfits or other similar work.

**Detroit Steam Traps, Tilting Tank Type, and Detroit Return Trap System** for boiler feed and water lift are products of the American Blower Co., Detroit, Mich., described in a newly-issued circular, including price lists and illustrations. Typical applications of the single trap and double trap systems as applied to boilers are illustrated. "ABC" Variable Speed Steam Engines for driving paper making machines are the subject of another interesting circular published by this company, the various types being illustrated in halftone engravings.

## New Firm to Handle Richmond Boilers and Radiators

**Cameron-Schroth-Cameron Co.,** Chicago, is the title of a new firm in the heating trade that has secured the exclusive selling rights of the Richmond line of boilers and radiators. It is composed of

W. A. CAMERON

W. A. Cameron, formerly treasurer of Kellogg-Mackay-Cameron Co., A. H. Schroth, formerly assistant secretary and western sales manager of the same company, and A. S. Cameron, formerly purchasing agent for Kellogg-Mackay-Cameron Co. The new company's field of operations will be the middle west and western states and, in addition to its boiler and radiator business, it will carry a full line of heating and plumbing supplies. The general offices and warehouse will be located at 189-193 Michigan Street, Chicago.

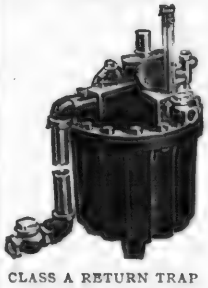
The Cameron-Schroth-Cameron Co. has organized with W. A. Cameron, pres-

A. H. SCHROTH

ident, A. H. Schroth, vice-president and A. S. Cameron secretary and treasurer. The company states in its announcement that the demand for Richmond boilers has confined the sales principally to the East, but with increased manufacturing facilities, the West will now be supplied and every effort will be made to make the name "Richmond" one of the best-known trade names in the western section of the country. The Richmond boiler is

A. S. CAMERON

one of the oldest on the market, having been manufactured for over 30 years. The firm emphasizes the statement that it will sell to the trade only.

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### Warren Webster & Company Extends Its Organization from Coast to Coast

Warren Webster & Co., Camden, N. J., announce that on and after January 1, 1909, the business heretofore carried on by the American Engineering Specialty Co., with headquarters at Chicago, and branches and agents in various cities through the middle west, will be conducted in the name of Warren Webster & Co., with main office and works at Camden, N. J. This change will give to architects, engineers and contractors, users and intending purchasers of "Webster" apparatus the full advantages of the "Webster" organization, which now covers all parts of the country. It implies no change in the personnel. Branch offices and representatives are located in the following cities: New York, Philadelphia, Boston, Atlanta, Pittsburg, Chicago, Cincinnati, Cleveland, Indianapolis, Minneapolis, Milwaukee, St. Louis, Kansas City, New Orleans, Denver, Salt Lake City, San Francisco, Los Angeles, Seattle, Portland, Montreal, with branches in various cities in Canada and London, Eng.

### New Buildings for the Department of Corrections, New York

**New York.**—Six prison buildings, aggregating in value $2,210,000, will be built for the Department of Corrections, New York City, on Riker's Island. They include a 3-story administration building to cost $400,000; a 4-story cell building for males to cost $1,200,000; a main penitentiary building, 4 stories high, to which will be annexed a cell building for females with a mess hall, the whole building to cost $350,000; and a separate mess hall and kitchen for the men, 1 story high, to cost $70,000. Trowbridge & Livingston, New York, are the architects, and Percival R. Moses, New York, has been selected as the designing engineer for the mechanical equipment.

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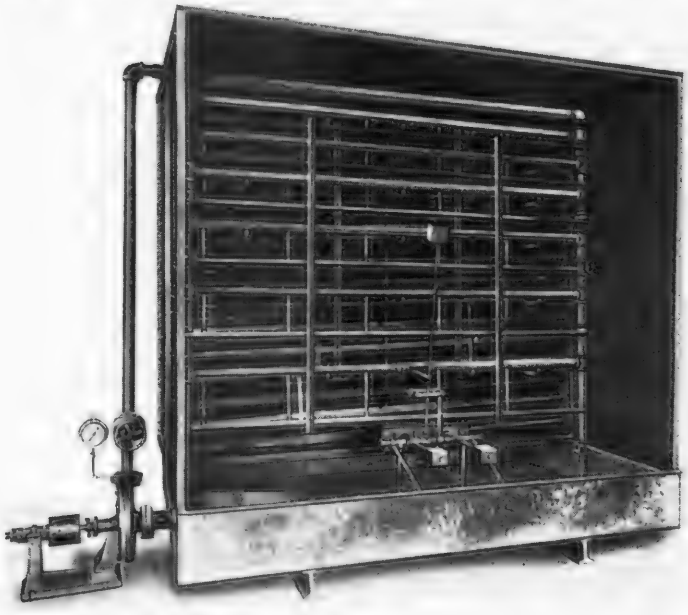

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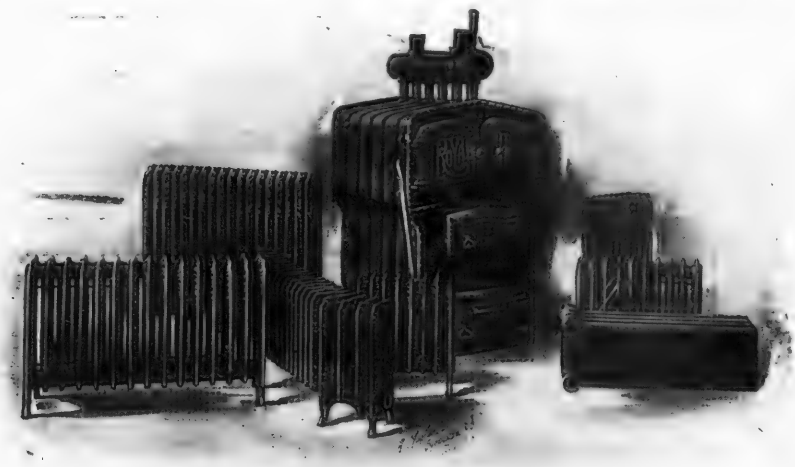

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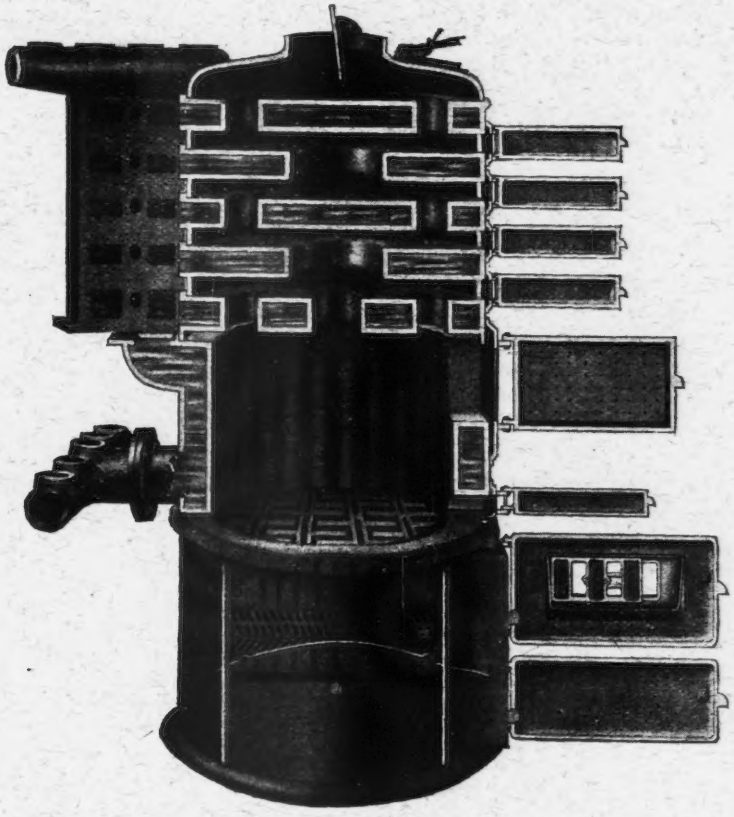

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